JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXV. MAY, 1913. No. 2.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY

(Under whose supervision this number is published).

Eng.-in-Chief R. S. GRIFFIN, U. S. N. Rear Admiral J. R. EDWARDS, U. S. N.

Eng.-in-Chief C. A. MCALLISTER, U. S. R. C. S. Lieut. Comdr. E. L. BENNETT, U. S. N.

Lieutenant ORMOND L. COX, U. S. N.

SCREW PROPELLERS.

METHOD OF DESIGN, DESIGN BY COMPARISON; EFFECT OF THRUST DEDUCTION AND REDUCTION OF DIAMETER ON THE PROPULSIVE COEFFICIENT.

BY CAPTAIN C. W. DYSON, U. S. NAVY, MEMBER.

INTRODUCTION.

In estimating the power required for the propulsion of any proposed hull, the Navy Department has practically abandoned all other methods than that of trials of the model of the proposed hull in the model tank, and from the results of these trials, obtains the curves of effective horsepowers based on speeds as abscissas. Other methods are used, however, as checks.

By the term "Effective Horsepower" is meant that power which when applied to a tow rope would pull the hull through the water at the desired speed.

The effective horsepower curves, as usually supplied to the Designing Branch of the Bureau, from which to estimate the necessary power for propulsion, are the curves of effective

horsepower for the bare hull only. Before the estimation of power can be made, the expected increase of effective horsepower due to the resistance of the appendages, or excrescences, as they are sometimes called, must be added to the bare hull power curves to obtain the total effective horsepower necessary.

These appendages consist of the following, the various items being given in the order in which they most frequently occur :

1. Rudder and Stern Post.
2. Bilge Keels.
3. Struts, Bosses and Shafting.
4. Dockings Keels.
5. Small Scoops over openings in hull.
6. Large Scoops over openings in hull.

No. 1 is encountered in all vessels, either single or multiple screw.

No. 2 is met with in most vessels of any considerable size.

No. 3 exists only in vessels having two or more propellers, although in some cases of single-screw vessels, the dead wood may be cut away and the propeller shaft supported by a strut. In some cases of twin-screw ships, the form of stern known as the "Lundborg" stern may be used and there will be no struts.

No. 4 is only met with in large, heavy vessels where such keels are required to better distribute the weight of the hull when docking.

No. 5 is found in all torpedo boats and destroyers built at the present date.

No. 6 is found in those torpedo boats and destroyers built from ten to twelve years ago.

The resistances due to the bilge and docking keels and shafts are probably those due to their wetted surfaces only and can be calculated as such. The other appendages enter into the total residual resistance (wave and eddy making),

and are estimated as varying according to Froude's Law of Comparison. That this law is correct when applied to such forms, is not certain, but it is adhered to for want of something more definite.

DERIVATION OF BLOCK COEFFICIENT TO USE IN CALCULATION OF PROPELLER.

Should we, for different classes of vessels, adopt standard sets of bow and stern lines and standard shapes of midship sections, we would have for all vessels of any class, no matter what the ratio of Beam to Length on the Load Water Line, an even condition of circumstances governing the flow of water to the propellers, and a change of percentage appendage resistance varying approximately directly as the ratio of Beam to this Length. We would find, however, that the nominal block coefficients (= 35 × Displacement ÷ Beam × Length on Load Water Line × Draught), would change nearly inversely with the beam ratio, while the actual body conditions, so far as affecting the propeller performance, would have remained the same.

Further, should two models of equal beam, length and displacement, but of different midship section forms, be towed, it would be found that the model having the finer midship section coefficient (= Area of midship section ÷ Beam × Draught) would require the greater tow-rope power. It would also be found that the longitudinal coefficient of this model (= 35 × Displacement ÷ Area of midship section × Length of Load Water Line) would be the larger of the two.

For two vessels of the same dimensions and for the same service, the appendage resistances would be the same, but if one vessel had a finer midship section than the other, the bare hull resistance of the former vessel would be greater than the latter, therefore the percentage appendage resistance would be less.

As the corrective Chart for Block Coefficient and the Chart for estimation of Appendage Resistance are closely related, it

will be well to explain them here and also explain how corrections for Block Coefficient are made when the location of the propellers differs from the location of the propellers on the Standard vessels whose propellers and results were used in the determination of the Charts of Propeller Design.

CHART I.—FOR CORRECTION OF BLOCK COEFFICIENT.

On this Chart the abscissas are Block Coefficients, while for the Block Correction, the ordinates are values of Beam ÷ Length on L.W.L. For checking the Block by means of the Coefficient of Immersed Midship Section, the ordinates are Coefficients of Immersed Midship Section. All vessels, whose Immersed Midship Section coincides with the Standard Curve, will be of standard fore and after body (abnormal designs of hull not being considered). Those plotting below the Standard Curve will be bluffer, forward and aft, and those plotting above the curve will be finer than the Standard Hulls, so that for vessels that plot below the Standard Curve, the percentage appendage resistance will be less, and for those that plot above, the percentage appendage resistance will be greater than Standard by the amount shown by the curves.

USE OF CHART I IN PROPELLER DESIGN.

In the lower section of this Chart are shown three diagonal lines, *X*, *Y* and *Z*. Line *X* is for the Standard vessels from which the Charts of propeller design were obtained. Such vessels have coefficients of Immersed Midship Section falling close to the curve of M.S. coefficient marked Standard, have propellers located well clear of the hull so that loss, through interference of flow of water to the screw by the hull, is a minimum and also the effect of the wake on the velocity of the flow of water to the propeller is a minimum.

As the location of the screws draws closer in behind the hull and the influence of the wake has sensibly increased over that of condition *X*, the line *Y* takes the place of *X*. Where the propeller is located close to and directly to the rear of the stern post, so that the full wake effect of the hull

is encountered, line *Z* replaces both *X* and *Y* in the determination of the Block Coefficient.

To apply this Chart, let us suppose we have three vessels, A, B and E, having the following characteristics:

	A.	B.	E.
Nominal Block Coef., ...	.78	.665	.61
Beam÷Length on L.W.L., ...	.123	.186	.217

It is required to find the Block Coefficient to use in the design of the propeller and also to estimate for the expected appendage resistances.

Plot *A*, *B* and *E* on the chart with the nominal block coefficients as abscissas and with the values of Beam÷L.L.W.L. as ordinates. Through these plotted points and the unity abscissa point, pass a straight line, extending it until it cuts line *X*. In the cases taken, *A*, *B* and *E* are all on the same line passing through the unity value of abscissas. Where this line crosses *X*, at *B*, project up to the Standard curve of midship section coefficient. Should the M.S. coefficient of the vessel in question plot near to the standard curve of M.S. coefficient, the vessel's ends may be considered standard, and the vessel's block coefficient be taken as that given by the abscissa value of *B*. Should it fall above this curve, that is, the M.S. be fuller than standard, the ends will be finer; if below, fuller than standard, and the block coefficients be modified accordingly. Should the propellers be located in the conditions given by lines *Y* or *Z* the fineness will be gaged, as before, by the intersection of the line with *X*, but the actual block to use for the propeller design will be that given by the abscissa value of the point of intersection with *Y* or *Z*, except where correction is made for variation from the Standard M.S. Coefficient.

CHART 7.—CORRECTION FOR THRUST DEDUCTION.

Another point to be considered, in the estimate of power and the design of the propeller, is the peculiar effect upon thrust of the propeller exercised by fullness of after-body

lines of the hull. This effect is not met with or is of such small extent as to be neglected, where the standard ship block coefficient to be used is .55 or less. As the standard ship block increases above .55, this evil effect becomes exaggerated and a corrective factor must be applied in designing the screw. This corrective factor is given by the Curves C, C^2 and C^3 on Chart 7, and is inserted in the equation for diameter, which is expressed thus:

$$\text{Diameter in feet} = \sqrt{\frac{2.88 \times \text{E.H.P.} \times C^3}{(\text{P.A.} \div \text{D.A.}) \times \text{E.T.} \times S}},$$

where E.H.P. = effective horsepower for the hull with appendages.

P.A. ÷ D.A. = Projected Area divided by Disc Area of screw = Projected Area Ratio.

S = Speed corresponding to the E.H.P.

E.T. = Effective thrust in lbs. per sq. in. on projected area of the propeller =

$$\frac{\text{E.H.P.} \times 33{,}000}{S \times 101.33 \times \text{Proj. Area in sq. in.}}$$

101.33 = Speed in feet per minute for a speed of one knot per hour.

The effect of this corrective factor is to increase diameter and pitch and decrease revolutions of the propeller. In most of the cases where it becomes necessary to use C, the diameters of the propellers that can be used are limited to a certain maximum size; this allowable size is usually smaller than the theoretical figures call for, so that for the same projected area as the propeller of full diameter, the projected area ratio of the actual screw is much increased. Further, by the use of C, the thrusts per square inch of projected area are very much lower than those of the Propeller Design Charts for equal tip speeds.

This lowering of thrusts is a direct result of the excessive value of the wake and of the thrust deduction in such vessels, and no attempt has ever been made heretofore, so far as the

writer is aware, to give a guide to the correction needed for these conditions of wake and thrust deduction.

The effect of reduction of diameter on propulsive coefficient will be shown later in discussing particular cases.

"THRUST DEDUCTION" ALSO CALLED "AUGMENT OF RESISTANCE."

A propeller driving a vessel acts very much after the nature of a pump, creating a suction column of water from forward to, and a pressure column of water aft from the screw. There is one difference in action to be noted, and that is the fact that the velocity of flow of the water in the suction column is less than it would be in a pump, due to the advance of the propeller itself along this column.

Should the propeller be so located in reference to the hull as to have the flow of water in this suction column seriously interfered with, so that the full supply of water which should flow, due to the suction head, is not given to the screw, a serious loss of propelling power will result.

The form of the hull, where the propeller is located in close proximity to it, may have such an adverse effect upon the screw as to make the obtaining of the designed speed an impossibility with the existing conditions. Such a case occurred a short time ago with two oil-fuel barges built for the Navy Department. These vessels were designed for a speed of six knots, and the effective horsepower curves indicated that this speed could be easily obtained with the power. However, after trying seven different propellers and developing a maximum I.H.P. of 176, the highest speed realized was only 5¼ knots.

The propeller had been located in the orthodox position during these trials, and the action of the water indicated that a portion of the feed was being drawn from astern. The line of shafting was then changed so as to lower the propeller about three feet, the lower blade projecting below the line of keel. In this new position, with propellers of the same pitch and surface but of six inches greater diameter, a speed of 6¼

knots was obtained with the same power. The greater portion of this increase in efficiency may be credited to the increase in diameter, and the remainder to the change in position.

Barnaby, in writing on this subject, makes the following statement: "If a screw is placed behind a stern so bluff that the supply of water is impeded, it will draw in water at the center of the driving face and throw it off from the tips of the blades like a centrifugal pump. It is recorded that an attempt to propel a square-ended caisson by means of a screw resulted in the caisson going astern, whichever way the screw was driven."

CHART 2.—ESTIMATE OF APPENDAGE RESISTANCE.

This subject is one of which little is known, and yet it is one of vital importance in the design of the propeller and the estimate of power required for propulsion, if anything like approximately accurate charts for design purposes are to be constructed.

For standard forms of forward and after bodies of the hull the writer has assumed that the appendage resistance percentages will vary directly as the ratio $\frac{B}{L.W.L.}$. Any error entering into this assumption is on the right side of the ledger and will result in a slightly higher estimate of resistance than actually exists.

Should the hull be finer than the standard the percentage resistance will increase above that given by Chart 2; should the hull be bluffer than standard the opposite effect will be produced, the amount of increase or decrease being indicated on Chart 1, curves of midship section coefficient.

On Chart 2 are drawn, with $\frac{B}{L.W.L.}$ as abscissas and values of percentage appendage resistance as ordinates, lines of percentage resistance for each appendage in the order in which they usually occur. Thus, the lowest line is for the rudder and rudder post, which are found on all vessels. The next line above, No. 2, is for the rudder, rudder post and bilge

keels; No. 5 is for the rudder, rudder post, bilge keels, shafts, bosses and struts, when carefully designed and placed; and finally, the line for total appendage resistance, No. 7, which, in addition to the appendages covered by No. 5, includes docking keels.

On the Chart is given a table of conditions governing the lines of resistance shown.

Should the model be towed with any of the appendages the percentage resistance due to these appendages should be subtracted from the total appendage resistance given by Chart 2 for all the appendages which the vessel under consideration is to carry, and the remaining percentage be applied to the E.H.P. given by the model-tank curve.

The usual method of judging of the performance of a propeller is to obtain the ratio $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ or $\frac{\text{E.H.P.}}{\text{S.H.P.}}$, and the higher these ratios the better are the propellers. Very often, and in fact more often than otherwise, the E.H.P. used in either of these ratios is for the bare hull only. This comparison, when the E.H.P. of the bare hull is used, is not just, and many a good propeller has been rated as a poor one on account of it. The real E.H.P. delivered by the propeller is that necessary for the hull with all appendages, and this is what should be used in estimating the propeller performance.

In order to better illustrate the handling of Charts 1, 2 and 7 the following two problems are submitted:

Problem (*a*).

Vessel having the ratio $\frac{\text{B}}{\text{L.L.W.L.}} = .200$;

Block Coefficient $= \frac{\text{Displacement} \times 35}{\text{L.L.W.L.} \times \text{Beam} \times \text{Draught}} = .525$;

Coefficient Immersed Midship Section $= .975$.

a. Twin Screws: (Appendages — Rudder, rudder post, shafts, struts and bosses, bilge keels.)

b. Single Screw: (Appendages—Rudder, rudder post and bilge keels.)

Find block coefficient for each case for use in propeller design, percentage appendage resistance and the thrust deduction factor *C*.

Solution.—Plot the position of the vessel on Chart 1, with the B.C. = .525 as abscissa and $\frac{B}{L.L.W.L.} = .2$ as ordinate, and through this plotted point and through the abscissa value of unity at *D* draw a straight line. This line crosses the standard line of B.C. at an abscissa value of .6, and the standard value of midship section coefficient corresponding to this abscissa value of .6 is .975, the same as that of the vessel; therefore the vessel's lines are regarded as standard. The value of B.C., therefore, for the twin-screw condition is .6, and for the single-screw condition will be at the intersection of the line drawn through the plotted position of the vessel and the line *Z*, for which the abscissa value = .72.

As the lines are standard no further correction than the above is necessary, and we take from Chart 2, line 5, abscissa value .2, the percentage correction for the appendages = 19.6 per cent. for the twin-screw arrangement, while for the single-screw arrangement this percentage = 4.6 per cent. is taken from Chart 2, line 2. For the thrust deduction factor *C* we enter Chart 7 with an abscissa value of .6 for which *C* = 1.04 for the twin-screw, and an abscissa value of .72 for which *C* = 1.23 for the single-screw arrangement.

Problem (*b*).—Vessel of the same ratio $\frac{B}{L.L.W.L} = .200$; same B.C. value = .525, but having a coefficient of immersed midship section .9. Two conditions as before, namely, twin and single screws with the same appendages as before. Find B.C., appendage percentage resistance and *C*.

Solution.—Plot the vessel's position on Chart 1 and draw the line through *D* as before. This line will intersect *X* and *Z* at abscissa values of .6 and .72 as before, but these are not the true block coefficients to use in obtaining slip values, as the standard midship section coefficient corresponding to abscissa value of .6 is .975, while the vessel in question has a

M.S. coefficient of .9. Assuming that the fullness of the vessel's lines will vary inversely as the M.S. coefficient we have

$$\text{Slip B.C.} = \text{Standard B.C.} \times \frac{\text{Standard M.S. Coefficient}}{\text{Actual M.S. Coefficient}};$$

$.6 \times \frac{.975}{.9} = .65$ for the twin-screw arrangement.

Drawing a line through the point on X having an abscissa value of .65, and through D, we find the intersection of this line with Z to have an abscissa value of .76, and this latter value will be the slip B.C. to use for the single screw. Referring again to Chart 1, Curves of M.S. coefficient, we see that for an abscissa value of .6 and M.S. coefficient of .9 the appendage resistance percentage is only 80 per cent. of the standard resistance given by Chart 2. By Problem 1 this was found to be 19.6 per cent. for the twin-screw and 4.6 per cent. for the single-screw arrangement; therefore, for the vessel of Problem 2 these percentages become $19.6 \times .80 = 15.68$ per cent. and $4.6 \times .80 = 3.68$ per cent. For the thrust deduction factors C we enter Chart 7 with abscissa values of .65 and .76 for the two conditions and the values obtained are for B.C. $= .65$, $C = 1.1$, and for B.C. $= .76$, $C = 1.31$.

The Screw Propeller.

THEORIES OF DESIGN.

The three most important theories of design, given in the order of their importance, are: 1. Froude's; 2. Rankine's, and 3. Greenhill's. The assumptions for each of these being as follows:

Froude.—Assumes the element as a small plane moving through the water along a line which makes a small angle with the direction of the plane. He then takes the normal pressure upon the elementary area, which gives propulsive effect to vary as the area, as the square of its speed and as the sine of the slip angle.

Rankine.—The fundamental assumption was that as the

propeller advanced with a certain slip all the water in an elementary ring of radius r was given a certain velocity in a direction perpendicular to the face of the blade at that radius. Then, from the principle of momentum, the thrust from the elementary ring is proportional to the quantity of water acted upon in one second, and to the sternward velocity communicated to it.

Greenhill.—Approaches the problem from a direction entirely different from that of either of the other two preceding theorists. He assumes that the propeller is working in a fixed tube with closed end. The result is that the motion transmitted to the water is wholly transverse. The blade is assumed perfectly smooth, so that the pressure produced by the reaction of the water is normal to the blade and has a fore-and-aft component which produces thrust.

In all the above theories, the loss by friction is taken as that due to the friction of the propelling plane moving edgewise, or nearly so, through the water.

In all of the theories in connection with which mathematical methods are to be used, it is practically necessary to regard the blade as having no thickness. This is a serious defect in all the theories, as they all use a true slip based upon true pitch and consider the designed pitch of the driving surface of the blade as this true pitch. The fact of the matter is that the face pitch of a blade with thickness, or its nominal pitch as it may be called, is very different from the true or actual pitch, and this fact causes complications in using the mathematical formulas.

It would be necessary, in case these theoretical formulas were adhered to, to compare each formula with experimental results and select that one which seemed to agree more closely. Then, using this as a semi-empirical formula, with coefficients and constants deduced from experiments or experience, problems could be satisfactorily dealt with. When the vast number of various conditions for which we may be called upon to design a propeller are considered, it is readily seen how impossible it would be to tabulate the correctors

which would be required to cover all, or even a large number of these conditions.

Casting aside the idea of designing propellers from purely theoretical formulas, as incorrect and impractical, there are then remaining for use the following

PRACTICAL METHODS OF DESIGN.

1. By "Direct Comparison," when all the conditions for a satisfactory vessel of a similar form to the one under consideration are known.

2. By methods based on trials of model propellers in model tanks. This may also be classed under the head of "direct comparison," as the Laws of Comparison are assumed to cover propellers as well as hulls.

3. By methods based on actual trials of full-sized propellers in service over carefully measured courses.

The first method practically insures a propeller of equal propulsive efficiency with that of the propeller on the compared vessel, but gives no opportunity for improvement in performance.

The second method is open to the decided objection that the conditions under which the model screw is tried in the tank are radically different from those under which the full-sized screw operates. In fact, propellers whose models have shown high tank efficiencies, have failed most signally in service, while other propellers whose models gave poor efficiency, have delivered a high propulsive coefficient. This latter has been ascribed to a high hull efficiency, but this explanation does not exactly satisfy when the fact is considered that where two or more propellers for the same vessel have been tested, that propeller whose model gave the highest efficiency has failed, while the propeller with the lower tank efficiency has succeeded.

The writer is inclined to the belief that the true cause of these discrepancies exists in the use of an incorrect method of derivation of the model screw dimensions from those of the full-size propeller.

METHOD OF "DESIGN BY COMPARISON."

The following orthodox method is taken from Peabody's Naval Architecture, and is that which is generally used:

D_1 = Diameter of original propeller.
D_2 = Diameter of 2d propeller or of model propeller.
L_1 = Length of compared vessel.
L_2 = Length of new vessel or model hull behind which model screw should operate if fitted to a hull (which is not done).
P_1 = Pitch of original propeller.
P_2 = Pitch of 2d or model propeller.
R_1 = Revolutions of original propeller.
R_2 = Revolutions of 2d or model propeller.
S_1 = Speed of compared vessels.
S_2 = Corresponding speed of new or model hull.

Then

$$D_2 = D_1 \times \frac{L_2}{L_1} = D_1 r;$$

$$R_2 = R_1 \left(\frac{L_1}{L_2}\right)^{\frac{1}{2}} = \frac{R_1}{r^{\frac{1}{2}}};$$

$$S_2 = S_1 \left(\frac{L_2}{L_1}\right)^{\frac{1}{2}} = S_1 r^{\frac{1}{2}};$$

$P_2 = p\,P_1 = r\,P_1$ where same ratio of $\frac{P}{D}$ is retained for the model as that of the original propeller.

$$\text{Apparent slip}_1 = \frac{P_1 \times R_1 - 101.33\,S_1}{P_1 \times R_1}.$$

$$\text{Apparent slip}_2 = \frac{pP_1 \times \frac{R_1}{r^{\frac{1}{2}}} - 101.33\,S_1 r^{\frac{1}{2}}}{pP_1 \times \frac{R_1}{r^{\frac{1}{2}}}} = \frac{pP_1 \times R_1 - 101.33\,S_1 r}{pP_1 \times R_1};$$

which, when $p = r$ becomes $\frac{P_1 \times R_1 - 101.33\ S_1}{P_1 \times R_1}$, or the apparent slip of the model, is equal to the apparent slip of the original propeller.

Tip speed$_1 = R_1 \times \pi D_1$;

Tip speed$_2 = R_2 \times \pi D_2 = \frac{R_1}{r^{\frac{1}{2}}} \times \pi D_1 r = R_1 \pi D_1 r^{\frac{1}{2}}$; and

$$T.S._2 \gtrless T.S._1 \text{ depending upon the value of } r.$$

Again,

$$I.H.P._2 = I.H.P._1 r^{\frac{1}{2}};$$

$$\text{Disc area}_2 = \tfrac{1}{4}\pi D_1^2 r^2;$$

$$\text{Disc area}_1 = \tfrac{1}{4}\pi D_1^2;$$

$$P_2 \times R_2 = \frac{p P_1 \times R_1}{r^{\frac{1}{2}}};$$

I.T. per sq. in. Disc Area$_2$

$$= \frac{I.H.P._1 \times r^{\frac{1}{2}} \times 33{,}000}{\frac{p P_1 \times R_1}{r^{\frac{1}{2}}} \times \frac{\pi}{4} D_1^2 r^2} = \frac{I.H.P._1 \times 132{,}000}{p P_1 \times R_1 \times \pi D_1^2};$$

I.T. per sq. in. Disc Area$_1$

$$= \frac{I.H.P._1 \times 33{,}000}{P_1 \times R_1 \times \frac{\pi}{4} D_1^2} = \frac{I.H.P._1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2};$$

$$\therefore \quad \frac{I.T._2}{I.T._1} = \frac{r^2}{p}, \text{ or where } p = r, \frac{I.T._2}{I.T._1} = r.$$

In other words, with different percentage losses from blade friction due to change in tip speeds, the model screw is supposed to deliver an equal percentage of the power driving it as effective thrust, with the original propeller, and its apparent slip is supposed to be equal to that of the original propeller, although the thrusts per square inch of disc area have been changed in the ratio r, the two screws working under approximately the same conditions of resistance.

In the method of comparison proposed by the writer, and which method, by the way, may be considered as highly unorthodox, the formulas take the following forms:

$$D_2 = D_1 r^{\frac{1}{3}};$$

$$P_2 = P_1 r^2;$$

$$R_2 = \frac{R_1}{r^{\frac{1}{3}}};$$

$$S_2 = S_1 r^{\frac{1}{3}};$$

$$\text{Apparent slip}_1 = \frac{P_1 \times R_1 - 101.33\, S_1}{P_1 \times R_1};$$

$$\text{Apparent slip}_2 =$$

$$= \frac{P_1 r^2 \times \frac{R_1}{r^{\frac{1}{3}}} - 101.33\, S_1 r^{\frac{1}{3}}}{P_1 r^2 \times \frac{R_1}{r^{\frac{1}{3}}}} = \frac{P_1 \times R_1 - 101.33\, S_1}{P_1 \times R_1}.$$

$\therefore$ Apparent slip_1 = Apparent slip_2;

$$\text{Tip Speed}_1 = R_1 \times \pi D_1;$$

$$\text{Tip Speed}_2 = R_2 \times \pi D_2 = \frac{R_1}{r^{\frac{1}{3}}} \times \pi D_1 r^{\frac{1}{3}} = R_1 \times \pi D_1.$$

$\therefore$ Tip Speed_1 = Tip Speed_2.

Again,

$$\text{I.H.P.}_2 = \text{I.H.P.}_1 r^{\frac{1}{3}};$$

$$\text{Disc Area}_2 = \frac{\pi}{4} D_1^2 r^{\frac{2}{3}};$$

$$\text{Disc Area}_1 = \frac{\pi}{4} D_1^2;$$

$$P_2 \times R_2 = P_1 r^2 \times \frac{R_1}{r^{\frac{1}{3}}} = P_1 r^{\frac{1}{3}} \times R_1;$$

I.T. per sq. in. Disc Area_2

$$= \frac{\text{I.H.P.}_1 \times r^{\frac{1}{3}} \times 33{,}000}{P_1 r^{\frac{1}{3}} \times R_1 \times \frac{\pi}{4} D_1^2 r^{\frac{2}{3}}} = \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2};$$

I.T. per sq. in. Disc Area_1

$$= \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2}.$$

$\therefore$ $I.T._2 = I.T._1$, and for the model screw the tip speed, apparent slip, and thrust per square inch of disc area are identical with those of the original screw. This identity existing in this case and not in the former, why should the latter not be taken as the proper method of comparison?

The method by which the latter method of comparison is derived will be explained later on.

THE THIRD "METHOD OF PRACTICAL DESIGN."

This method is open to the criticism that results are so easily affected by inaccuracies in timing runs over the measured course, by inaccurate count of revolutions and inaccurate power measurements, by variation of conditions as to smoothness of immersed body of the ship and surfaces of the propellers, variations in trim and displacement of the vessel from the model conditions, adverse conditions of wind and sea, and variable skill of helmsmen.

These variables may, with proper care, be reduced so as to have the least possible effect, and, when this care is taken, the results obtained are quite as good for designing purposes, if not better, than can be obtained by any other method.

In conformity with the foregoing statement, the writer has developed a series of Charts for use in propeller design, which, up to date, have given uniformly good results when basic hull data of sufficient accuracy has been at hand. A brief description of the development of the method and of the charts will now be given.

DEVELOPMENT OF THE METHOD.

In Froude's investigations of the action of screw propellers and in Barnaby's development of Froude's results, the model propellers used had a constant form of developed area of blade and a constant ratio of greatest width of development to the diameter of the screw.

The form of developed area used was an ellipse of which the major axis was the radius of the propeller, and this form was used for all values of pitch ratio (P ÷ D) that were experimented with.

The constants obtained by Barnaby are expressed by the following formulas:

$$C_A = \frac{A.V.^3}{I.H.P.};$$

$$C_R = \frac{R \times D}{V}, \text{ in which}$$

A = Disc area of the propeller (including hub), in square feet.

R = Revolutions per minute of the propeller.

D = Diameter of the propeller, in feet.

I.H.P. = Indicated horsepower applied on the screw = 2 × effective horsepower or tow-rope horsepower required for the bare hull at the speed S.

V = Speed of the water through the screw, in knots per hour.

For convenience in computation, in the above formulas, the speed of the water through the propeller was replaced by the speed S of the ship in knots per hour, an assumed standard wake of ten per cent. being applied. A standard thrust deduction of ten per cent. was also adopted, as representing the addition to tow-rope resistance caused by the action of the propeller on the stream lines of the ship.

Assuming that for propellers identical in all respects except that of developed area, the thrust that can be delivered will vary directly as the developed area, Barnaby's constants can be represented graphically as a chart, and on this chart can be plotted from their performances, the location of any desired propeller.

In developing the writer's method of design, many propellers were so plotted, and from the results were obtained a sufficient number of factors to enable those factors of propeller

design, which directly effect the efficiency, to be so tied together graphically, as to render it possible to take from these charts, when the resistance and form of hull are known, the necessary factors for use in the determination of the diameter, pitch, projected area, and probable propulsive coefficient that will be obtained, for any desired case.

These charts are shown marked 3, 4, 5, 6 and 7, and the method of using them will be explained later.

STANDARD FORMS OF PROJECTED AREAS OF BLADES FOR USE WITH THE CHARTS OF DESIGN.

Returning to the Barnaby presentation of Froude's results, the constants obtained by Barnaby are only correct so long as the ratio of developed area to disc area and the elliptical form of this developed area used by him, are adhered to. There is no way of allowing for the effect of increase or decrease in this developed area except the rough one of estimating that the total thrust that can be delivered by the propeller will vary directly as the developed area.

By investigation of what occurs when the standard elliptical blade used by Barnaby in his experiments is broadened or narrowed, it is readily seen that this method of correcting for variations in developed area ratio is incorrect, for as the blade widens, for any one pitch ratio, the length of its resistance arm increases above that of the standard width blade, and as it narrows, the length of this arm decreases. In the first case, the resistance of the blade to turning is increased not only by the increased surface friction of the larger blade but also by the increase in the length of the radius to the center of pressure of the blade. Should the blade be narrowed below standard, the opposite effect will be produced.

If elliptical blades of varying pitch ratios, but having the same area of projection on the disc, be laid down, it will be noted that the form of projection, not only on the disc but also on the plane of the axis of the propeller, changes as we

pass from the lower to the higher pitch ratios. This same change in form of projected area also occurs if we lay down blades of the same pitch ratio but of different values of developed area ratio, thus in both cases showing that not only the resistance of the blade due to change of surface, and in the first case change of pitch, has been affected, but it has been still further modified by the change in the distribution of this surface and the modification in the leverage arm of the blade resistance.

A distinguished educator who, until lately, was an advocate for the use of constant developed area form in design work, has lately put before the public a work on propellers in which he advocates the use of the projected area in place of developed area. He has adopted, instead of the constant elliptical form of development, a constant elliptical form of projected area. Should the projected area ratio be .3 its form is that of an ellipse, and should this ratio be .6 the form is again an ellipse. The same changes in distribution of surface occur as before, and no benefit has been obtained except that of having an easy form to lay down and also one that can be mathematically represented.

In order to maintain as constant a distribution of blade area as possible and thus guard against changes in resistance due to changes in distribution the writer adopted as a basic form of projected area the form of projection of the blades of two propellers having approximately a projected area ratio of .32. These two propellers had most excellent records, which we could hardly hope to better.

Using a standard hub diameter equal to .2 the diameter of the propeller, this standard basic form of projection was drawn. Then, with the center of the hub as a center, and different radii, circular arcs were struck crossing the axis of the projection. In obtaining the projected forms for areas differing from the basic area the widths of the projections measured on these circular arcs were made proportional to the circular arc widths of the basic projection; that is, a .6

projected area ratio would have circular arc measurements $\frac{60}{32}$ times as great as those for the basic .32 projection.

The forms of projected area so obtained, when compared with the forms of blades of many propellers, are found to agree very closely, from the lowest to the highest values of P.A. ÷ D.A., with those forms which have the best records credited to them.

By using these forms, for any pitch ratio the resistance arm of the blade always remains the same, no matter what the developed or projected area ratio, and the only change in resistance to turning is that caused by the additional surface. The forms of projection, both on the disc and on the fore-and-aft plane, remain constant for all pitch ratios.

Naval Constructor D. W. Taylor, in writing on the effect of blade form as deduced from model tank experiments, stated "A good practical rule would seem to be to make the blades broader at the tip for low pitch ratios and narrow them for high ones." The blades here advocated follow this rule automatically, as the broadest part of the developed blade, measured on the elliptical development of the circular projected arc, moves slowly in towards the hub as the pitch ratio increases, thus gradually narrowing the tip of the blade for the higher pitch ratios.

The derived projected forms are shown on Plate 8, where is also shown a chart by which the developed area ratio can be obtained for standard propellers of any pitch ratio and any given projected area ratio, and *vice versa*. On this same chart sheet is given a table of multipliers for obtaining the length of half-chord widths for different projected-area-ratio blades for any desired diameter of propeller, by means of which the projected-area forms can be laid down without the the use of the chart of forms.

FORM OF BLADE SECTIONS FOR STANDARD BLADES.

In the propellers designed according to the writer's method, the form of section existing in the propellers from whose data

the charts were developed, has been adhered to. In these blades, the working face of the blade forms the nominal pitch surface, the blades in all cases being made with constant pitch. The thickness of the blade is built up on the back of this nominal pitch face. The form of the back is an arc of a circle and the edges are made as thin and sharp as possible without sacrificing durability to an extravagant degree.

Should the thickness, instead of all being applied to the back of the pitch surface, be divided on each side of it, the thrusts exerted by the blade per revolution will decrease and the tip speeds will increase, until the metal is equally divided on each side of the pitch face when the nominal pitch of the blade becomes equal to the true pitch. That this distribution of metal results in this equalization of true and nominal pitches is demonstrated by the fact that if a model blade of zero nominal pitch be revolved in water and the thrusts measured, that blade having the metal divided equally on each side of the pitch plane will develop zero thrust, while all other blades will register thrust.

Should the leading side of the pitch face of a blade be thrown back in an attempt to fine up the entering edge the true pitch will be increased to a still greater extent over the nominal pitch, the speed of the blade will be lowered and the thrusts will increase. Should the after side of the blade be drawn back from the pitch plane there will, in the opinion of the writer, be a decided tendency toward the production of cavities in the wake of the blade, with resultant vibration.

EFFECT OF CHANGE OF BLADE FORM.

Should the forms of projected areas here advocated not be adhered to, the following results may be confidently looked for:

1. Broadening the blades at the tips.—Revolutions will be decreased, apparent slip will be decreased and thrusts will be increased.

2. Narrowing the blades at the tips.—Revolutions will be increased, apparent slip increased and thrusts decreased.

In the matter of relative weights for equal blade strengths the narrow-tipped blade has the advantage.

The writer wishes it to be distinctly understood that no claim is made that the forms advocated by him are necessarily those giving the maximum efficiency. He is of the opinion that equal efficiencies can be obtained with all shapes, if for each shape the proper diameter, pitch and surface have been provided for the absorption of the delivered power under the conditions in which the screw is operating. Each series of forms must, however, have its own particular factors of design if results in conformity with the computed performance are to be expected.

THICKNESS OF THE BLADE.

The fiber stress to be used in determining the thickness of the blade at the root depends upon the material of which the blade is to be made and the degree of approximation of the point of design to full overload conditions. The material usually used for propellers in the Naval Service is manganese bronze, and the specified ultimate tensile strength of the material is 60,000 pounds. Where the possible overload does not exceed ten per cent. a working stress of 10,000 pounds per square inch can be used with safety. This is the condition existing for Chart 5 of the Charts of Design. For propellers designed by Chart 6, where the power used in the calculations may be very much lower than the maximum power possible, this working stress should be reduced to about 6,000 pounds.

The formula used for the determination of blade thickness has been derived from Naval Constructor D. W. Taylor's work on "Resistance of Ships and Screw Propulsion," and is an adaptation of the formula proposed by him. The nomenclature and formulas are as follows:

T = Thickness of blade at and tangent to hub, additional thickness due to fillets being neglected. T should not exceed .2 W.

W = Width of blade tangent to hub.

A = (33,000 × I.H.P.) ÷ 2π × Revolutions × Number of blades = 5,252 I.H.P. ÷ R × N = Maximum indicated torque per blade, in foot pounds.

B = .31 × Diameter of screw, in feet = Mean arm.

C = A ÷ B = Resultant athwartship force on one blade, in foot pounds.

D = 12 × B — Radius of hub, in inches = Arm of athwartship force measured to root of blade.

E = C × D = Athwartship moment at root of blade, in inch pounds.

F = (33,000 × I.H.P.) ÷ Pitch, in feet × Revolutions × Number of blades = Indicated thrust per blade, in pounds.

G = .345 × Diameter of propeller, in inches = Mean arm of thrust, in inches.

H = G — Radius of hub, in inches = Arm of thrust measured to root of blade, in inches.

J = F × H = Fore-and-aft moment at root of blade, in inch pounds.

K = Circumference of hub, in feet ÷ Pitch, in feet = Tangent of angle between face of blade and center line of hub or fore-and-aft line tangent to surface of hub.

L = Sine of arc whose tangent is K.

M = Cosine of arc whose tangent is K.

N = L × J = Component of fore-and-aft moment normal to face of blade at root.

O = M × E = Same for athwartship moment.

P = N + O = Total moment at root of blade in inch pounds.

f = Fiber stress = $\begin{pmatrix} 10{,}000 \text{ for chart } 5 \\ 6{,}000 \text{ for chart } 6 \end{pmatrix}$ for manganese bronze of 60,000 T.S.

$$T = \sqrt{\frac{P \times 13.125}{W \times f}}.$$

CHARTS OF DESIGN.

These charts are five in number and are marked as numbers 3, 4, 5, 6 and 7.

Chart 3 is the Chart of Propulsive coefficients; 4 that of Tip Speeds. Although the lines of design from which Charts 5 and 6 are obtained are shown on Charts 3 and 4, these two latter charts are not used for actual design work but are used after the propeller has been designed, when it becomes desirable to lay down curves of expected performance through a range of speeds.

The actual Charts of Design are but two in number, namely 5 and 6. Before explaining these charts, it is advisable to explain the meanings of the different terms used in the charts.

DEFINITIONS OF TERMS AND ABBREVIATIONS.

I.H.P. = Indicated Horsepower of Main Engine on one screw.

S.H.P. = B.H.P. = .92 I.H.P. = Shaft or Brake Horsepower applied to the line shafting, and measured by torsion of shaft abaft the thrust bearing.

E.H.P. = Effective or Tow-rope Horsepower required to tow the hull at any given speed, the hull being fitted with all its appendages

$$= \frac{\text{Total E.H.P.}}{\text{Number of propellers}}.$$

P.C. = Actual Propulsive Coefficient

$$= \text{P.C.}_1 \div \text{C} = \text{P.C.}_1 \times \frac{1}{\text{C}} \left(\frac{\text{A}^8 - s_1}{\text{A}^8 (1 - s_1)} \right)^3.$$

P.C._1 = Propulsive Coefficient of the Propeller (Chart)

$$= \text{C} \times \frac{\text{E.H.P.}}{\text{I.H.P.}} = \text{C} \times \frac{\text{E.H.P.} \times .92}{\text{S.H.P.}}.$$

C = Thrust Deduction Factor for Standard Block Coefficient, from Chart 7.

D.A. = Disc area of propeller, and includes the area of the hub.

P.A. = Projected area of the propeller blades.

P.A. ÷ D.A. = Projected area ratio.

H.A. = Helicoidal (developed) area of the propeller blades.

H.A. ÷ D.A. = Developed area ratio.

P = Pitch of the propeller, in feet = Feet advance per revolution of the propeller if working in a non-yielding medium.

D = Diameter of the propeller, in feet.

R = Revolutions per minute of the propeller.

S = Speed of ship, in knots per hour.

s = Apparent slip of screw, expressed as a decimal $= [(P \times R) - (S \times 101.33)] \div P \times R$.

Wake veloc. = Rate of motion of water directly at stern of vessel in same direction as the vessel is moving.

T.S. $= \pi \times R \times D$ = Tip speed of the screw, in feet per minute.

Total Indicated Thrust on one screw

$$= \frac{I.H.P. \times 33{,}000}{P \times R} = I.T. = \frac{S.H.P. \times 33{,}000}{.92 \times P \times R}.$$

Total Propulsive Thrust on one screw.

$$= P.T. = I.T. \times P.C._1.$$

Total Speed Thrust

$$= \frac{I.H.P. \times 33{,}000}{S \times 101.33} = S.T. = \frac{S.H.P. \times 33{,}000}{.92 \times S \times 101.33}.$$

Total Effective Thrust on one screw

$$= E.T. = S.T. \times P.C._1.$$

$$\frac{I.T.}{S.T.} = \frac{I.H.P. \times 33{,}000}{P \times R}$$

$$\times \frac{S \times 101.33}{I.H.P. \times 33{,}000} = \frac{S \times 101.33}{P \times R} = 1 - s.$$

$$\frac{P.T.}{E.T.} = \frac{I.T. \times P.C._1}{S.T. \times P.C._1} = \frac{I.T.}{S.T.} = 1 - s.$$

$$I.T._d = \frac{I.T.}{\frac{\pi}{4} \times D^2 \times 144} = \text{Indicated thrust per square inch of disc area.}$$

$$I.T._p = \frac{I.T.}{P.A. \times 144} = \frac{I.T.}{\frac{P.A.}{D.A.} \times \frac{\pi}{4} D^2 \times 144} = \text{Indicat'd thrust per square inch of projected area.}$$

$P.T._p = I.T._p \times P.C._1 =$ Propulsive thrust per square inch of projected area.

$E.T._p = P.T._p \div (1 - s) =$ Effective thrust per square inch of projected area.

The formulas to use in calculation of the propeller coefficients take the following forms:

$$(1)\ D = \left.\sqrt{\frac{291.8 \times I.H.P \times C^2}{P \times R \times I.T._d}}\right\} \text{Full diameter.} = \left.\sqrt{\frac{291.8 \times I.H.P. \times C^2 \times A^4}{P \times R \times I.T._d}}\right\} \text{Limited diameter.}$$

$$(2)\ D = \left.\sqrt{\frac{2.88 \times E.H.P. \times C.^3}{\frac{P.A.}{D.A.} \times E.T._p \times S}}\right\} \text{Full diameter.} = \left.\sqrt{\frac{2.88 \times E.H.P. \times C^3 \times A^4}{\frac{P.A._1}{D.A._1} \times E.T._p \times S_1}}\right\} \text{Limited diameter.}$$

$$P \times R = \frac{101.33 \times E.T._p}{P.T._p} \qquad (3)$$

$$P = \frac{101.33\, S \times E.T._p \times \pi D}{T.S. \times P.T._p} = \frac{P \times R \times \pi D}{T.S.} \qquad (4)$$

$$R = \frac{T.S.}{\pi D}$$

The charts as constructed, are based on a mechanical efficiency, for reciprocating engines of .92, and the propulsive coefficients only fit this efficiency. The relation between the I.H.P., S.H.P., and E.H.P. being expressed by $I.H.P. = \frac{S.H.P.}{.92.} = \frac{C \times E.H.P.}{P.C.}$, should the mechanical efficiency differ from .92, the relation between these powers must be corrected

accordingly, thus suppose a mechanical efficiency of only .85 is expected, then I.H.P. $= \frac{\text{S.H.P.}}{.85} = \frac{\text{C} \times \text{E.H.P.} \times .92}{\text{P.C.} \times .85}$, and the I.H.P. to use in equation (1) would be only $\frac{85}{92} \times$ this final I.H.P. of the main engines.

NUMBER OF BLADES.

The propeller charts have been developed from the performances of three-bladed propellers and therefore a correction must be applied should it be desired to use any other number of blades than three.

Should a four-blade wheel be required, multiply the I.H.P. of equation (1) or the E.H.P. of equation (2) by .865 and use the product in place of the actual I.H.P. or E.H.P. in calculating for the values of D, P and R. Multiply the projected area corresponding to these values of D, P and R by 4/3 to obtain the projected area of the four-bladed wheel.

Should we desire a two-bladed wheel, divide the I.H.P. or E.H.P. by .75 and multiply the projected area by $\frac{3}{2}$.

PROPELLERS OF REDUCED DIAMETER.

In many cases of propeller design, after computing a propeller to absorb a certain amount of power at a given number of revolutions, it will be found that, due to restrictions as to draught and hull clearance, the deduced diameter cannot be carried. It then becomes necessary to calculate a secondary screw of less diameter to absorb the designed power without exceeding the designed revolutions.

An investigation of the performances of propellers where the diameters, after preliminary trials, have been slightly reduced, accompanied by a slight reduction in pitch, shows, by comparison of performances before and after reduction, that where equal powers are applied to the screws the

$$\frac{\text{Revolutions after reduction}}{\text{Revolutions before reduction}} = \sqrt{\frac{\text{Diameter before reduction}}{\text{Diameter after reduction}}};$$

$$\text{or } \frac{R_2}{R_1} = \left(\frac{D_1}{D_2}\right)^{\frac{1}{2}} \text{ and that}$$

$$\frac{\text{Apparent slip after reduction}}{\text{Apparent slip before reduction}} = \frac{s_2}{s_1} = \left(\frac{D_1}{D_2}\right)^4.$$

The tip speeds after and before reduction will have the following ratios:

$$\frac{\text{T.S.}_2}{\text{T.S.}_1} = A = \frac{\pi R_2 D_2}{\pi R_1 D_1} = \frac{D_2}{D_1} \times \left(\frac{D_1}{D_2}\right)^{\frac{1}{2}} = \left(\frac{D_2}{D_1}\right)^{\frac{1}{2}};$$

$$\therefore \quad D_2 = D_1 \times A^2, \text{ and } A = A^2 \times \frac{R_2}{R_1}; \therefore R_2 = R_1 \div A.$$

Assuming that the total projected areas are the same in the two cases, which can be done safely, as the primary screws used were narrow tipped so that the reduction in area due to cutting the diameters was negligible, and also, as a slight correction has been allowed in the revolution and slip exponents to allow for the slight increase in the area of the secondary screw necessary to bring it equal to that of the primary one, the equation for projected area ratios takes the following form:

$$\text{P.A.}_2 \div \text{D.A.}_2 \ : \ \text{P.A}_1 \div \text{D.A.}_1 \ :: \ D_1^2 \ : \ D_2^2;$$

$$\text{P.A.}_2 \div \text{D.A.}_2 = \text{P.A.}_1 \div \text{D.A.}_1 \times \left(\frac{D_1}{D_2}\right)^2 = \text{P.A.}_1 \div \text{D.A.}_1 \times \frac{1}{A^4}.$$

The apparent slips and the pitches of the primary and secondary propellers will bear the following ratios to each other:

$$\frac{s_2}{s_1} = \left(\frac{D_1}{D_2}\right)^4 = \frac{1}{A^8}; \therefore \ s_2 = \frac{s_1}{A^8} \text{ and } P_2 = P_1 \times A.$$

DESCRIPTION OF THE CHARTS.

Chart 3. Chart of Propulsive Coefficients.—The ordinates of this chart are P.C.'s expressed as percentages, and the abscissas are pounds indicated thrust per square inch of disc

area. The curves of P.C. are given for different values of P.A. ÷ D.A. and cover the practical ranges of this ratio.

It will be noted that all of these curves come tangent to an enveloping curve and then fall rapidly. Coinciding very closely with this enveloping curve is a curve for design which will be found again on Chart 5. Crossing these curves of P.C. at their points of maximum efficiency, is shown another curve which will be found reproduced on Chart 6 as a curve of design.

Chart 4. Curves of Tip Speeds.—The ordinates of this chart are tip speeds in feet per minute, while the abscissas are, as on Chart 3, pounds indicated thrust per square inch of disc area. On this chart as on Chart 3, the curves of tip speeds are shown for different values of P.A. ÷ D.A., and cover all practical values of that ratio. The curves are all tangent to an enveloping curve, as in Chart 3, and are crossed by two design lines of tip speeds, corresponding to the cross curves of design shown on Chart 3, and which are again shown on Design Charts 5 and 6.

These two charts, 3 and 4, are only used when laying down estimated curves of performance of a vessel. The data for design of propellers and estimate of power required are obtained by the use of either Chart 5 or 6, depending on the conditions under which the propeller operates.

Chart 5. Chart of Design.—High tip speeds, high speed of ship, maximum speed and power of vessel much greater than ordinary cruising speed and power.

Chart 6. Chart of Design.—Low tip speeds, low to moderate speed of ship, maximum speed of vessel not much greater than ordinary continuous speed, maximum power considerably in excess of power for continuous speed, this latter power and speed being used in the design. We may, however, design a propeller by each chart and then interpolate between for the final screw.

Propellers designed by Chart 5, have their points of maximum efficiency at considerably lower thrusts than at the point of design, while those designed by Chart 6 have their

points of maximum efficiency coincident with the points of design.

Chart 5 is applicable to vessels where the power applied to the propellers is high, Chart 6 being used for those of low powers. In no case, however, should the propeller from Chart 5 be used when the resultant P.A. ratio is much less than .25.

The ordinates of these charts are P.C. in percentage; pounds thrust, $I.T._d$, $I.T._p$, $P.T._p$, and $E.T._p$; and tip speeds in feet per minute. The abscissas are values of P.A.÷D.A.

Commencing at the top of each chart, the upper curve is the design curve of P.C.; the second curve is a curve of $I.T._p$, which, however, is not used in the calculations; the third is a series of curves of $E.T._p$ for standard block coefficients ranging from .2 to .9; the fourth is a curve of $P.T._p$; the fifth of tip speeds; and finally, the sixth curve is that of $I.T._d$.

To obtain data from these curves, suppose the vessel under consideration should have a block coefficient, standard by Chart 1, of .6, and we choose to enter the chart, say Chart 5, with a P.A.÷D.A. of .3; the data to be used in the calculations will be on the ordinate having the abscissa value of .3, and is as follows:

P.C. = .677, $E.T._p$ (from .6 curve) = 9.78, $P.T._p$ = 8.45, T.S. = 6,650, and should we desire to use equation (1) for the diameter, $I.T._d$ = 3.74. The value of C, the thrust deduction factor, is obtained for Standard B.C. = .6, from Chart 7. C = 1.04.

The apparent slip will equal $1 - \frac{P.T._p}{E.T._p} = 1 - \frac{8.45}{9.78} = .136.$

Should we use Chart 6, the data would become P.C. = .6935, $E.T._p$ = 6.4, $P.T._p$ = 5.79, T.S. = 5,600, $I.T._d$ = 2.5, and the apparent slip would equal $1 - \frac{5.79}{6.4} = .0953.$

REMARKS ON CHARTS 5, 6 AND 7.

An examination of these charts will show at once that they tie together, in a consistent manner, all the elements necessary to be taken into account in the design of a propeller,

namely, the form of the ship with its influence on wake, flow of water to the screw and resultant slips, the effective and propulsive thrusts, the indicated thrusts and the rotary speed of the screw through the water. As any one of these elements is changed a corresponding change takes place in all the others to meet it. All are regarded as of equal importance.

While the values of P.A. ÷ D.A. have been taken for abscissas it is not because it is considered of more importance than the other elements, for such is not the case, as an examination of the charts reveals the fact that up to a value of .3 all projected area ratios deliver nearly equal propulsive coefficients. Beyond .3, however, the propulsive coefficient falls rapidly with an increase of this ratio. This equality of propulsive coefficient for a wide variation of P.A. ÷ D.A. is often claimed, and it is seen that up to this value of .3 it is justified.

Later it will be shown how the value of P.A. ÷ D.A. varies in passing from Chart 6 to Chart 5, for the same conditions, and how little variation of P.C. results.

EFFECT OF CHANGE IN CHARACTERISTICS OF PROPELLER.

Suppose the propeller to be designed with blades separate from the hub. Call the designed pitch P, the diameter D and the blade area A, what will occur if we should change any one or more of these characteristics?

1. Pitch and blade area constant. Increase D. Effect: Efficiency increases, slip decreases, revolutions decrease. Decreasing D will have the opposite effect.

2. D and A remain constant. Increase P. Effect: Revolutions will decrease, slip will increase, efficiency will be increased at low thrusts and decreased at high ones. If P should be decreased, the opposite effects will result.

3. P and D remain constant. Increase A. Effect: Revolutions decrease, slip decreases, little effect on propulsive efficiency.

Finally, as a guide to judge the effect of general changes in the characteristics, should we take a constant form of hull, a constant E.H.P., and a constant speed, and should develop a series of propellers of different projected area ratios for these conditions, we would find that as we passed from the lowest to the highest projected area ratios, the diameter would decrease, the pitch would decrease, the revolutions would increase and the slips would increase.

The mistake is often made of supposing that a propeller is efficient because its apparent slip is low. This mistake is only justified to the extent that small projected area ratios have lower slips and have higher propulsive coefficients than have the higher projected area ratios, but the slips for a block coefficient of .2 will be considerably greater than for a block coefficient of .9, the projected area ratios and propulsive coefficients being the same for both blocks.

Thus, for P.A. ÷ D.A. of .25 and of .6, for B.C. = .2 and .9, the apparent slips would be as follows, from Chart 5:

B.C.	P.A. ÷ D.A. = .25	P.A. ÷ D.A = .6
.2	.2019	.2552
.9	.0589	.1256

PROBLEMS MET WITH IN PROPELLER DESIGN.

These may be divided into two general classes: 1st, Problem of full diameter, and 2d, Problems of reduced diameter. The first are usually solved by the use of Chart 5, and the second by either 5 or 6, or by interpolation between them. These classes of problems may be still further divided into problems of sufficient and problems of insufficient data.

Two forms of computation are all that are needed, however, as where insufficient data is given, the missing data is not filled in and the propeller obtained is a guess at the best, and the client, if the work is being done for outside parties, should be so informed and no responsibility for results should be accepted.

In problems of insufficient data, the usual data supplied are as follows:

S = Estimated speed.
I.H.P. = Designed indicated horsepower of engines.
N = Number of propellers.
R = Designed revolutions for I.H.P. and S.
B.C. = Block coefficient of vessel.
$\frac{B}{L.W.L.}$ is not usually given but can generally be obtained.
Appendages fitted can usually be ascertained.

In problems of sufficient data, the following are the available data:

S = Designed speed.
E.H.P. = Effective horsepower (bare hull) for S.
I.H.P. = Designed power of engines [may be given or I.H.P. to give E.H.P. + Appendages) may be obtained from work].
N = Number of propellers fitted.
R = Designed revolutions for designed I.H.P.
B.C. = Block coefficient.
$\frac{B}{L.W.L.}$ = Ratio of greatest immersed beam to length on load water line.
I.M.S. = Coefficient of immersed midship section.

FORMS FOR COMPUTATION.

Each form should have at the top the name of the vessel, the name of the computer, the date and the available data, with the form of computation following; the number of the design chart used being also stated.

Form 1. Problems of Reduced Diameter.—Vessels to be designed for a constant cruising speed S. Diameter to be limited not to exceed a certain amount and revolutions fixed at a certain number. Usually Chart 6, except when abnormally large P.A. ÷ D.A. results.

(1) D = Maximum allowable diameter.

(2) R = Designed revolutions.

(3) $T.S. = \pi RD$ = Tip speed of secondary screw.

(4) $T.S._1$ = Tip speed of primary screw. Must equal or be greater than T.S.

(5) $A = T.S. \div T.S._1$.

(6) A^4.

(7) A_8.

(8) $P.A._1 \div D.A._1$ = Projected area ratio corresponding to $T.S._1$ by Chart.

(9) $P.A. \div D.A. = (P.A._1 \div D.A._1) \div A^4$ = Projected area ratio.

(10) $I.T._d$ = Indicated thrust per square inch of disc area corresponding to $P.A._1 \div D.A._1$ by Chart.

(11) $P.T._p$ = Propulsive thrust corresponding to $P.A._1 \div D.A._1$ by Chart.

(12) $E.T._p$ = Effective thrust corresponding to $P.A._1 \div D.A._1$ and corrected block coefficient by Chart.

(13) $$\frac{P.T._p}{E.T._p} = 1 - s_1.$$

(14) $$s_1 = 1 - \frac{P.T._p}{E.T._p} = \text{Apparent slip of primary screw.}$$

(15) $$s = \frac{s_1}{A^8} = \text{Apparent slip of secondary screw.}$$

(16) $1 - s = 1 -$ Apparent slip of secondary screw, in decimal.

(17) S = Designed speed of vessel, in knots per hour.

(18) $S \times 101.33$ = Speed of vessel, in feet per minute.

(19) $$P \times R = P_1 \times R_1 = \frac{S \times 101.33}{1 - s} = \text{Pitch} \times \text{Revolutions of both primary and secondary screws.}$$

(20) $$S_1 = \frac{P_1 \times R_1 \times (1 - s_1)}{101.33} = \text{Speed to use in formulas.}$$

(21) $$\text{E.H.P.} = \text{Effective horsepower for speed S.}$$

(22) $$\text{E.H.P.}_1 = \text{E.H.P.} \times \left(\frac{S_1}{S}\right)^3 = \text{Estimated effective horsepower for } S_1.$$

(23) $$C = \text{Thrust deduction factor for standard block from Chart 7.}$$

(24) $$\text{P.C.}_1 = \text{Propulsive coefficient for P.A.}_1 \div \text{D.A.}_1 \text{ from Chart.}$$

(25) $$\text{I.H.P.} = C \times \frac{\text{E.H.P.}_1}{\text{P.C.}_1} = \text{Est. power required.}$$

(26) $$\text{P.C.} = \frac{\text{E.H.P.}}{\text{I.H.P.}} = \text{Actual est. propulsive coeffi.}$$

(27) $$D = \sqrt{\frac{2.88 \times \text{E.H.P.}_1 \times C^3 \times A^4}{\frac{\text{P.A.}_1}{\text{D.A.}_1} \times \text{E.T.}_p \times S_1}}$$

$$= \sqrt{\frac{2.88 \times \text{E.H.P.}_1 \times C^3}{\frac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p \times S_1}}$$

$$= \sqrt{\frac{291.8 \times \text{I.H.P.} \times C^2 \times A^4}{P_1 \times R_1 \times \text{I.T.}_d}}.$$

(28) $$P = \frac{P \times R \times \pi D}{\text{T.S.}} = \text{Pitch.}$$

(29) $$R = \frac{\text{T.S.}}{\pi D} = \text{Revolutions.}$$

Should a four-bladed propeller be required, use .865 E.H.P._1, or .865 I.H.P. in the equation for D, and for P.A. ÷ D.A. multiply equation (9) by 4/3.

Form 2. Problems of Full Diameter. Chart 5.

(1) P.A. ÷ D.A. = Projected area ratio from Chart.

(2) T.S. = Tip speed for P.A. ÷ D.A.

(3) I.D._d = Indicated thrust per square inch of disc area corresponding to P.A. ÷ D.A.

(4) $P.T._p$ = Propulsive thrust per square inch of projected area corresponding to P.A. ÷ D.A.

(5) $E.T._p$ = Effective thrust for standard B.C. for P.A. ÷ D.A.

(6) $1 - s = \frac{P.T._p}{E.T._d} = 1 -$ Apparent slip expressed as a decimal.

(7) S = Designed speed, in knots per hour.

(8) $S \times 101.33$ = Designed speed, in feet per minute.

(9) $P \times R = \frac{S \times 101.33}{1 - s}$ = Pitch × revolutions of the propeller.

(10) E.H.P. = Effective horsepower (all appendages) for speed S.

(11) C = Thrust deduction factor for standard block coefficient from Chart 7.

(12) $P.C._1$ = Propulsive coefficient from chart for P.A. ÷ D.A.

(13) $I.H.P. = C \times \frac{E.H.P.}{P.C._1}$ = Estimated I.H.P. for speed S.

(14) S.H.P. = I.H.P. × .92 = Estimated shaft horsepower for speed S.

(15) $$D = \sqrt{\frac{2.88 \times E.H.P. \times C^3}{\frac{P.A.}{D.A.} \times E.T._p \times S}}$$

$$= \sqrt{\frac{291.8 \times I.H.P. \times C^2}{P \times R \times I.T._d}} = \text{Diameter of propeller.}$$

(16) $P = \frac{P \times R \times \pi D}{T.S.}$ = Pitch of propeller.

(17) $R = \frac{T.S.}{\pi D}$ = Revolutions of propeller.

$$(18) \qquad P.C. = \frac{E.H.P.}{I.H.P.} = \frac{P.C._1}{C}.$$

For a 4-bladed propeller correct E.H.P., I.H.P. and P.A. ÷ D.A. as before.

Should the problem be one of insufficient data, with either form, use the second formula for D, namely:

$$D = \sqrt{\frac{291.8 \times I.H.P. \times C^2}{P \times R \times I.T._d}}$$ for full diameter, or

$$D = \sqrt{\frac{291.8 \times I.H.P. \times C^2 \times A^4}{P \times R \times I.T._d}}$$ for reduced diameter.

See table of Forms at end of article.

PROBLEMS IN DESIGN AND ESTIMATING.

Problem 1. Full Diameter. Maximum Thrust. Chart 5.

Battleship, speed 21 knots. E.H.P. (bare hull) = 16,600.

Length on load water line = 600 feet. Beam = 97 feet 6 inches.

$\frac{B}{L.W.L.} = .1625$. Nominal block coefficient = .654.

Standard block coefficient = .625 (Chart 1).

Appendage resistance (two-shaft) = .176; (four-shaft) = .15 (Chart 2).

Coefficient M.S. = Standard.

Total resistance (two-shaft) = 16,600 × 1.176 = 19,528.

E.H.P. on one shaft = 9,764.

Total resistance (four-shaft) = 16,600 × 1.15 = 19,090.

E.H.P. on one shaft = 4,773.

C = 1.06; C^3 = 1.2.

Revolutions desired for two-shaft arrangement = 111; four-shaft = 220.

For two-shaft arrangement assume values of P.A. ÷ D.A. of .30, .31, .32, .33 and .34.

For four-shaft arrangement assume values of P.A. ÷ D.A. of .400, .425, .450, .475 and .500.

Use Form No. 3 of Formulas for Computation.

Two-Shaft.

P.A. ÷ D.A.....	.30	.31	.32	.33	.34
T.S., feet.....	6650	6880	7080	7280	7470
P.T.$_p$, pounds.....	8.45	8.57	8.68	8.78	8.87
E.T.$_p$, pounds.....	9.72	9.90	10.03	10.15	10.30
$1-s$	.8693	.8656	.8656	.8656	.8612
E.H.P., 1 shaft....	9764	9764	9764	9764	9764
C × E.H.P.....	10350	10350	10350	10350	10350
P.C.	.6775	.6730	.6680	.6630	.6575
2 × I.H.P.....	30554	30758	30988	31222	31498
S.....	21	21	21	21	21
101.33 S	2127.93	2127.93	2127.93	2127.93	2127.93
P × R.....	2447.9	2458.4	2458.4	2458.4	2470.9
Diameter, feet.....	23.39	22.80	22.29	21.82	21.34
Pitch, feet.....	27.04	25.59	24.32	23.15	22.18
R.p.m.....	90.51	96.07	101.1	106.2	111.4
2.88.....log	0.45939	...	...	...	...
E.H.P. =.....log	3.98963	...	...	...	...
C^3.....log	0.07593	...	...	...	...
(1)..... .....log	4.52495	4.52495	4.52495	4.52495	4.52495
P.A. + D.A. ...log	9.47712	9.49136	9.50515	9.51851	9.53148
E.T.....log	0.98767	0.99564	1.00130	1.00647	1.01284
S .....log	1.32222	1.32222	1.32222	1.32222	1.32222
(2)..... .....log	1.78701	1.80922	1.82867	1.84720	1.86654
D^2=(1)—(2)..log	2.73794	2.71573	2.69628	2.67775	2.65841
D..... .....log	1.36897	1.35787	1.34814	1.33888	1.32921
π .....log	0.49715	0.49715	0.49715	0.49715	0.49715
π D = (3).....log	1.86612	1.85502	1.84529	1.83603	1.82636
P × R.....log	3.38879	3.39065	3.39065	3.39065	3.39285
P × R × π D...log	5.25491	5.24567	5.23594	5.22668	5.21921
T.S. = (4).....log	3.82282	3.83759	3.85003	3.86213	3.87332
P..... ....log	1.43209	1.40808	1.38591	1.36455	4.34589
R=(4)—(3)...log	1.95670	1.98257	2.00474	2.02610	2.04696

Propeller for 111 Revolutions.

D = 21′ — 5″.

P = 22′ — 3″.

P.A. ÷ D.A. = .3392.

R.P.M. = 111.

I.H.P. = 31,475.

Four-Shaft.

P.A. ÷ D.A.	.400	.425	.45	.475	.500
T.S.	8550	9000	9500	10000	10550
P.T.$_p$	9.30	9.43	9.55	9.67	9.78
E.T.$_p$	10.88	11.05	11.23	11.41	11.61
$1-s$	.8548	.8534	.8504	.8475	.8424
E.H.P	4773	4773	4773	4773	4773
C × E.H.P	5060	5060	5060	5060	5060
P.C	.618	.602	.588	.573	.558
I.H.P	8188	8405	8606	8831	9068
S.H.P. (total)=I.H.P. × 4 × .92 =	30132	30930	31670	32518	33370
S	21	21	21	21	21
101.33 × S	2127.93	2127.93	2127.93	2127.93	2127.93
P × R	2489.4	2493.5	2502.3	2510.8	2514.2
D	13.38	12.96	12.42	11.99	11.59
P	12.24	11.28	10.28	9.46	8.68
R	203.3	221	243.5	265.4	289.8

E.H.P log	3.67879	...	...	...	...
2.88 log	0.45939	...	...	...	...
C^3 log	0.07593	...	...	...	...
(1) log	4.21411	4.21411	4.21411	4.21411	4.21411
P.A. ÷ D.A. log	9.60206	9.62839	9.65321	9.67669	9.69897
E.T. log	1.03663	1.03822	1.05038	1.05729	1.06483
S log	1.32222	1.32222	1.32222	1.32222	1.32222
(2) log	1.96091	1.98883	2.02581	2.05620	2.08602
D^2=(1) − (2) log	2.25320	2.22528	2.18830	2.15791	2.12809
D log	1.12660	1.11264	1.09415	1.07896	1.06405
π log	0.49715	0.49715	0.49715	0.49715	0.49715
πD = (3) log	1.62375	1.60979	1.59130	1.57611	1.56120
P × R log	3.39609	3.39681	3.39834	3.39981	3.40040
P × R × πD log	5.01984	5.00660	4.98964	4.97592	4.96160
T.S = (4) log	3.93197	3.95424	3.97772	4.00000	4.02325
P log	1.08787	1.05326	1.01192	0.97592	0.93835
R = (4) − (3) log	2.30822	2.34445	2.38642	2.42389	2.46205

Propeller for 220 Revolutions.

D = 12′ 11¼″.
P = 11′ 4½″.
P.A. ÷ D.A. = .4215.
R.P.M. = 220.
S.H.P. = 30,800.

Having obtained the propellers and the powers required to obtain the designed speed, it becomes necessary to lay down the "Curves of Performance" which we anticipate, that is, curves of "Speed—Power" and "Speed—Revolutions," as from these curves are obtained the expected fuel and water consumptions of the vessel for different speeds.

The charts to be used in obtaining these curves are Nos. 3 and 4, while the formulas are as follows:

T.S. = Tip speed for any thrust $I.T._d$ for P.A. ÷ D.A. of propeller, Chart 4.

$I.T._d$ = Indicated thrust per square inch disc area for T.S., Chart 4.

P.C. = Propulsive coefficient corresponding to P.A. ÷ D.A. and $I.T._d$, Chart 3.

C = Thrust deduction factor from Chart 7, corresponding to standard B.C.

P = Pitch of screw.

D = Diameter of screw.

R = Revolutions of screw per minute $= \frac{T.S.}{\pi D}$.

I.H.P. = Indicated horsepower for $I.T._d$ on one screw.

S.H.P. = .92 I.H.P.

E.H.P. = Effective horsepower for $I.T._d$.

$$E.H.P. = \frac{I.H.P. \times P.C.}{C}; \quad \text{Total E.H.P.} = n \times E.H.P.$$

$$R = \frac{T.S.}{\pi D}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$I.H.P. = \frac{P \times R \times D^2 \times I.T._d}{291.8 \times C^2} = \frac{P \times T.S. \times D \times I.T._d}{291.8 \times C^2 \times \pi}$$

$$= \frac{P \times T.S. \times D \times I.T._d}{916.7 \times C^2}, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

n = Number of shafts.

$n \times$ I.H.P. = Total I.H.P.; $n \times$ S.H.P. $= .92 \times$ Total I.H.P.

S = Speed corresponding to $n \times$ E.H.P. as obtained from model tank curves of "E.H.P.—Speed."

The work necessary to obtain the curves of performance for the two foregoing conditions follows:

Condition 1—Reciprocating Engines—Two Shafts. P.A. ÷ D.A. = .3392.

I.T.d.	T.S., Chart 4.	Pitch, feet.	Diameter, feet.	C.	I.H.P., Equation 2.	2 × I.H.P.	P.C., Chart 3.	$\frac{2 \times \text{E.H.P.}}{2 \times \text{I.H.P.} \times \text{P.C.}} = \frac{}{C}$	Speed.	R.P.M.	Final P.C. = P.C. + C.
4.6	7450	22.25	21.4	1.06	15735	31470	.6577	19528	21	111	.6205
4.0	7100	22.25	21.4	1.06	13130	26260	.6725	16660	20.38	105.6	.6344
3.5	6760	22.25	21.4	1.06	10940	21880	.6820	14080	19.6	100.6	.6434
3.0	6350	22.25	21.4	1.06	8806	17612	.6860	11400	18.36	94.45	.6472
2.5	5900	22.25	21.4	1.06	6818	13636	.6870	8838	17.0	87.76	.6481
2.0	5300	22.25	21.4	1.06	4900	9800	.6850	6333	15.4	78.84	.6462
1.5	4500	22.25	21.4	1.06	3120	6240	.6825	4017	13.32	66.93	.6438
1.0	3350	22.25	21.4	1.06	1549	3098	.6810	1990	10.8	49.83	.6424

Condition 2.—Turbine Engines.—Four Shafts.

P.A. ÷ D.A. = .4215.

I.T.d.	T.S., Chart 4.	Pitch, feet.	Diameter, feet.	C.	S.H.P. = Equation 2 × .92.	4 × S.H.P.	P.C., Chart 3.	$\frac{4 \times \text{E.H.P.} = 4 \times}{\text{S.H.P.} \times \text{P.C.}} = \frac{}{.92\ C.}$	Speed.	R.P.M.	Final P.C. = P.C. + C.
6.5	8950	11.38	12.94	1.06	7700	30800	.6025	19090	21	220	.6340
6.0	8780	11.38	12.94	1.06	6929	27716	.6240	17732	20.7	216	.6398
5.5	8530	11.38	12.94	1.06	6170	24680	.6375	16136	20.33	209.8	.6537
5.0	8300	11.38	12.94	1.06	5458	21832	.6500	14556	19.89	204.2	.6665
4.5	8000	11.38	12.94	1.06	4735	18940	.6590	12800	19.18	196.8	.6758
4.0	7710	11.38	12.94	1.06	4056	16224	.6626	11024	18.3	189.7	.6795
3.5	7350	11.38	12.94	1.06	3384	13536	.6650	9228	17.32	180.8	.6819
3.0	6920	11.38	12.94	1.06	2731	10924	.6640	7436	16.28	170.2	.6809
2.5	6350	11.38	12.94	1.06	2088	8352	.6615	5664	14.95	156.2	.6783
2.0	5550	11.38	12.94	1.06	1460	5840	.6575	3937	13.32	136.5	.6742
1.5	4500	11.38	12.94	1.06	888	3552	.6525	2376	11.4	110.7	.6691
1.0	3250	11.38	12.94	1.06	428	1712	.6480	1136	9.2	79.95	.6645

The Speed-Revolutions, Speed-I.H.P., and Speed-S.H.P. curves obtained above are shown on Fig. 1.

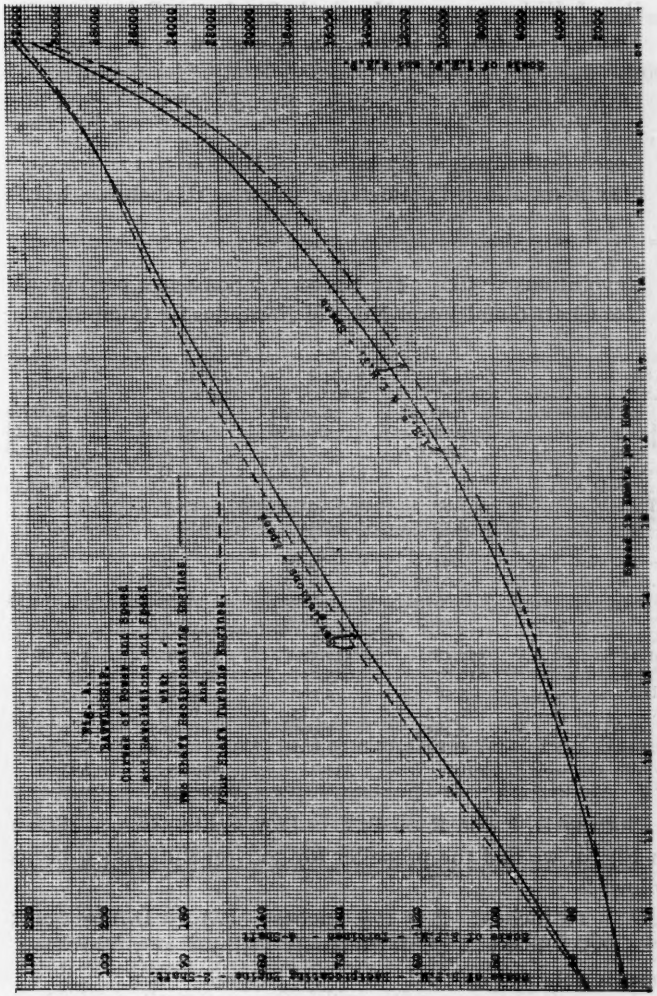


FIG. 1.

Problem 2. Reduced Diameter. Insufficient Data.

Motor Dory.—L.L.W.L. = 18.4 feet. B = 5 feet 1 inch. B ÷ L.W.L. = .277. Nominal Block Coefficient = .467. Draught = 1 foot $2\frac{1}{4}$ inches. Single screw in wake of stern post.

Displacement = 3,320 pounds. Standard B.C. = .76 (Chart 1, line Z).

Speed (estimated) = 5.7 knots. S.H.P. = 5.117.

Maximum allowed diameter of screw = 16 inches. I.H.P. $= \frac{5.117}{.92} = 5.562$.

Revolutions = 500. Tip Speed = 2,094.4 feet per minute. Use Chart 6 and Form No. 2 of Formulas for Computation.

T.S. ...	2,094.4	2,094.4	2,094.4	2,094.4
$T.S._1$	2,094.4	2,200.0	2,400.0	2,600.0
A (log)	0.00000	9.97864	9.94082	9.90612
A^4 (log)	0.00000	9.91456	9.76328	9.62448
A^8 (log)	0.00000	9.82912	9.52656	9.24896
$P.A._1 \div D.A._1$	.142	.1475	.1575	.167
P.A. ÷ D.A.	.142	.180	.2717	.3965
$I.T._d$	.88	.93	1.03	1.13
$P.T._p$	4.33	4.44	4.58	4.73
$E.T._p$	4.58	4.68	4.84	5.00
$1 - s_1$	.9455	.9487	.9464	.9460
s_1	.0545	.0513	.0536	.0540
s	.0545	.07603	.15944	.30439
$1 - s$	.9455	.92397	.84056	.69561
S	5.7	5.7	5.7	5.7
101.33 S	577.58	577.58	577.58	577.58
P × R	610.87	625.11	687.13	830.32
I.H.P.	5.562	5.562	5.562	5.562
C	1.313	1.313	1.313	1.313
D (feet)	2.2854	1.9918	1.5167	1.1227
P (feet)	2.0941	1.8677	1.5632	1.3983
R.p.m.	291.7	334.7	439.55	593.85

Plotting D, P and R on P.A. ÷ D.A. as abscissas, as shown on Fig. 2, we find the propeller for 500 revolutions to be as follows:

D = 16 inches; P = $17\frac{3}{4}$ inches; P.A. ÷ D.A. = .323; R = 500; Speed = 5.7 (estimated); Slip = 21.7 per cent. Results obtained on trial: R = 558; Speed = 6.26.

The slips with propellers of reduced diameter are always

higher than would be the case if the full diameter screw could be used. In some cases, where the decrease in diameter is very great, the slip may be nearly 50 per cent., and the existence of such an amount of slip should not be taken as

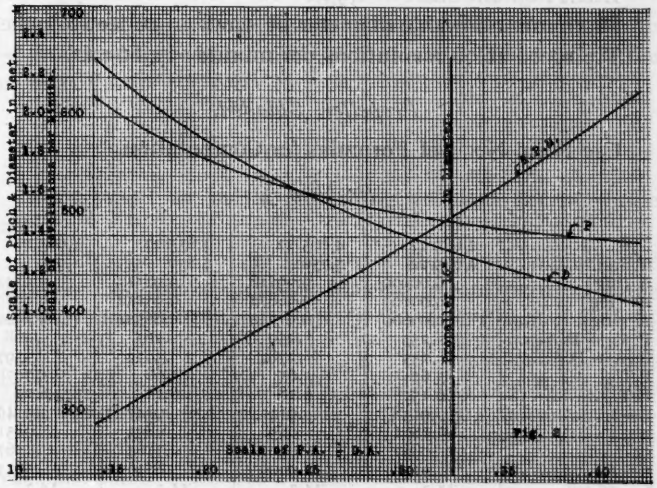


FIG. 2.

an accurate guide by which to condemn the propeller, for the propeller may be the best that can be fitted under the conditions existing, and the blame be laid on the conditions and not on the screw.

Problem 3. Problem of Reduced Diameter. Full Data.

Collier—Guaranteed continuous speed = 14.3 knots. Use Chart 6. Revolutions for this speed, 90 per minute.

L.W.L. = 528 feet; $\frac{B}{L.W.L.}$ = .1235; Nominal Block Coefficient = .726.

B = 65 feet $2\frac{3}{4}$ inches; Standard B.C. (Chart 1) = .615 (Line X).

Midship Section = .984; Twin Screw. E.H.P. (bare hull) = 3,430 for 14.3 knots.

Appendages { Struts and shafts. / Rudder and post. / Bilge keels.

E.H.P. (all appendages, Chart 2) = 3,430 × 1.129 = 3,872.

E.H.P. on one shaft = 1,936.

Maximum allowable diameter of screw = 16 feet 6 inches.

Tip speed at 90 revolutions per minute = 4,665.22.

Propeller, 3-bladed.

Use Form No. 1 of Formulas for Computation.

Problem 3.

T.S.	4,665.22	4,665.22	4,655.22	4,665.22
T.S.$_1$	4,665.22	4,750	4,850	4,950
A (log)	0.00000	9.99218	9.98313	9.97426
A^4 (log)	0.00000	9.96872	9.93252	9.89704
A^8 (log)	0.00000	9.93744	9.86504	9.79408
P.A.$_1$ ÷ D.A.$_1$	.255	.259	.263	.268
P.A. ÷ D.A.	.255	.2784	.3072	.3397
P.T.$_p$	5.62	5.65	5.68	5.70
E.T.$_p$	6.15	6.18	6.21	6.23
$1 - s_1$	.9138	.9142	.9147	.9149
s_1	.0862	.0858	.0853	.0851
s	.0862	.0991	.1164	.1367
$1 - s$	.9138	.9009	.8836	.8633
S	14.3	14.3	14.3	14.3
101.33 S	1,449	1,449	1,449	1,449
P × R	1,585.74	1,608.4	1,640	1,678.5
S_1	14.30	14.511	14.803	15.155
E.H.P.	1,936	1,936	1,936	1,936
E.H.P.$_1$	1,936	2,023.2	2,147.4	2,304.5
C	1.055	1.055	1.055	1.055
$EHP_{11} = C \times EHP_1$	2,042.5	2,134.5	2,265.5	2,431.2
P.C.$_1$	.699	.6985	.698	.6977
I.H.P.	5,844	6,112	6,491.3	6,969.2
P.C.	.6626	.6335	.5965	.5556
D (feet)	17.086	16.556	16.036	15.588
P (feet)	18.246	17.932	17.709	17.62
R.p.m.	86.91	89.70	92.60	95.26

Plotting these results of D, P and R on P.A. ÷ D.A. as abscissas, we find the propellers to have the following characteristics:

D = 16 feet 6 inches;

P = 17 feet 11 inches;

P.A. ÷ D.A. = .281;

I.H.P. = 6,150.

It is usually considered that greater efficiency of propulsion can be obtained by the use of a single propeller. Whether this is true or not depends entirely upon the revolutions chosen and the diameter that can be carried, as is shown by the following problem where the conditions for the single screw are exaggerated in order to show the large differences that can be produced by bad choice of revolutions. The vessel is the same as that of the preceding problem.

Problem 4. Reduced Diameter. Sufficient Data.

Same vessel as in the preceding problem except single screw.

Revolutions = 50 per minute. Maximum diameter of screw = 27 feet 4 inches. Maximum tip speed = 4,288 feet per minute. Standard B. C. (Line Z, Chart 1) = .735. E.H.P (Bare hull) = 3,450. C = 1.26, 4-bladed screw. E.H.P. (All appendages) Chart 2 = 3,450 × 1.029 = 3,550. (Use Chart 5.)

T.S.	4,288.	4,288.	4,288.	4,288.
$T.S._1$	4,288.	4,400.	4,500.	4,600.
A (log)	0.00000	9.98880	9.97904	9.96949
A^4 (log)	0.00000	9.95520	9.91616	9.87796
A^8 (log)	0.00000	9.91040	9.83232	9.75592
$P.A._1 \div D.A._1$	.204	.208	.212	.215
P.A. ÷ D.A.	.204	.2306	.2571	.2847
$P.T._p$	6.67	6.80	6.87	6.94
$E.T._p$	7.35	7.5	7.6	7.7
$1 - s_1$	.9075	.9067	.9040	.9013
s_1	.0925	.0933	.0960	.0987
s	.0925	.1147	.1383	.1735
$1 - s$	.9075	.8853	.8617	.8265
S	14.3	14.3	14.3	14.3
101.33 × S	1,449.0	1,449.0	1,449.0	1,449.0
P × R	1,596.7	1,636.77	1,681.58	1,753.24
S_1	14.3	14.646	15.002	15.594
.865 × E.H.P.	3,071.	3,071.	3,071.	3,071.
.865 × $E.H.P._1$	3,071.0	3,299.0	3,545.75	3,982.64
C	1.26	1.26	1.26	1.26
$P.C._1$	.6995	.6992	.6988	.6985
I.H.P	6,395.	6,873.	7,391.	8,305.
D (feet)	28.87	26.53	25.52	25.73
P (feet)	33.77	33.01	32.68	33.05
R.P.M.	47.28	49.58	51.46	53.04
$P.A._{11} \div D.A._{11}$	.272	.3075	.3428	.3796

Plotting these points, we find the propeller necessary, has the following characteristics:

D = 27.33 feet.
P = 32.9 feet.
P.A. ÷ D.A. = .315.
I.H.P. = 6,975.
E.H.P. = 3,550.
S = 14.3 knots.
P.C. = 50.9 per cent.

Comparing the propulsive coefficients obtained by the use of twin and single propellers in the foregoing two examples, we see that we obtain with

	P.C. (All Append.)	P.C. (Bare hull).
Single Screw, per cent.	50.9	49.46
Twin Screws, per cent.	62.96	56.11

Should a higher number of revolutions be chosen for the single screw and the reduction in diameter be less, these differences between the P.C.'s would be decreased. Should we be able to use the full diameter (primary), in both cases, there is no reason why equal P.C.'s on the hull with all appendages could not be obtained, and in such case, the single screw would be decidedly the better commercial proposition. This matter of choice of number of propellers will be discussed later.

Problem 5. Limitation of Speed due to Limitation of Propeller Diameter.

There were built for the U. S. Navy in 1912 two fuel-oil barges of the following dimensions: L.L.W.L. = 160 feet; Greatest Immersed Beam=25 feet; Draught, mean=8 feet 3 inches; Displacement = 835.5 tons.

The designed speed was six knots, and to obtain this speed a steam engine of 175 I.H.P. at 200 revolutions, driving a single screw, was fitted. As originally installed, a diameter

of 6 feet 3 inches was the maximum that could be carried, and with a diameter of 6 feet 2 inches and a 4-bladed screw, the maximum speed obtained was 5.258 knots, the I.H.P. being 160, and the revolutions 207.6. The pitch of this propeller was 3 feet 6 inches.

Let us see what Chart 6 shows as to the speed possibility under these same condition.

Data.

L.L.W.L. = 160 feet; $\frac{B}{L.W.L.} = .15625$; Standard B.C. = .895; C = 1.59; B = 25 feet.

Draught = 8 feet 3 inches; Nominal B. C. = .886.

Displacement = 835.5 tons.

There are available E.H.P. curves from which to obtain data.

T.S.	4,022.	4,022.	4,022.	4,022.
T.S 1	4,400.	4,800.	5,200.	5,600.
A (log)	9.96099	9.92320	9.88844	9.85625
A^4 (log)	9.84396	9.69280	9.55376	9.42500
A^8 (log)	9.68792	9.38560	9.10752	8.85000
P.A.$_1$ ÷ D.A.$_1$	.243	.261	.28	.3
P.A. ÷ D.A.	.3481	.5418	.7822	1.1272
I.T.$_d$	1.93	2.13	2.31	2.52
P.T.$_p$	5.55	5.66	5.73	5.78
E.T.$_p$	5.73	5.85	5.93	5.99
$1 - s_1$	.9686	.9675	.9663	.9655
s_1	.0314	.0325	.0337	.0345
s	.06442	.1337	.2631	.4873
$1 - s$	.93558	.8663	.7369	.5127
I.H.P.	160.	160.	160.	160.
P.C.$_1$	.70	.6985	.697	.694
E.H.P.$_{11}$=I.H.P. × P.C.$_1$	112.	111.76	111.52	111.04
C	1.59	1.59	1.59	1.59
E.H.P.$_1$ = E.H.P.$_{11}$÷C	70.44	70.3	70.13	69.8
S_1 (Corresponding to E.H.P.$_1$)	6.32	6.31	6.30	6.29
Log (S_1×101.33)	2.80646	2.80577	2.80508	2.80439
Log (P × R)	2.82032	2.82012	2.81997	2.81964
S	6.105	5.65	4.805	3.34
D (feet)	7.474	5.98	4.893	4.041
P (feet)	3.86	3.087	2.525	2.084
R.P.M.	171.3	214.1	261.6	316.8
P.A.$_{11}$ ÷ D.A.$_{11}$	.4641	.7224	1.043	1.503

Plotting these results, we find that the propeller should have the following dimensions, as given under the heading "Chart Propeller," but that six knots could not be made with this Chart Propeller.

	Chart Propeller.	Actual Propeller.
D = ...	6′ 2″	6′ 2″
P = ...	3′ 3″	3′ 6″
P. A. ÷ D.A. = . .	.678	.446
R = ...	207.6	207.6
S=	5.75	5.248
I.H.P. = . . .	160.	160.
P.C. = . . .	35.6 per cent.	26.55 per ct.

The pitch of the actual screw was too high and its area too small for the conditions existing and the diameter fitted. Having failed on trial, the line of shafting was lowered, permitting the fitting of a propeller of 6 feet 9 inches diameter. With this new condition, the following results were obtained: I.H.P.=177.64; Speed=6.235; Revolutions =206.27. The propeller was 3-bladed. The chart shows us as follows; Problem 6.

T.S.....	4,375.	4,375.	4,375.	4,375.
$T.S._1$.....	4,500.	4,800.	5,000.	5,200.
A (log).....	9.98777	9.95974	9.94201	9.92498
A^4 (log).....	9.95108	9.83896	9.76804	9.69942
A^8 (log).....	9.90216	9.67792	9.53608	9.39984
$P.A._1 \div D.A._1$.....	.248	.261	.271	.28
P.A.÷D.A.....	.2776	.3782	.4623	.5588
$P.T._p$.....	5.58	5.66	5.70	5.73
$E.T._p$.....	5.76	5.85	5.89	5.93
$1-s_1$.....	.9687	.9675	.9670	.9663
s_1.....	.0313	.0325	.0330	.0337
s.....	.0392	.0682	.0960	.1342
$1-s$.....	.9608	.9318	.9040	.8658
S.....	6.235	6.235	6.235	6.235
log (S×101.33)....	2.80058	2.80058	2.80058	2.80058
log (P×R).....	2.81795	2.83126	2.84441	2.86316
S_1.....	6.286	6.474	6.67	6.95
$E.H.P._1$ (from E.H.P. curve)..	69.5	73.1	76.8	82.78
C.....	1.59	1.59	1.59	1.59
$P.C._1$.....	.6995	.6985	.6975	.6965
I.H.P.....	158.	166.4	175.	189.
D (feet).....	8.947	7.687	6.997	6.447
P (feet).....	4.225	3.743	3.511	3.378
R.P.M.....	155.6	181.2	199.0	216.0

For Problem 6, the propeller will be as follows:

	Chart Propeller.	Actual Propeller.
D=	6′ 9″	6′ 9″
P=	3′ 5½″	3′ 6″
P.A.÷D.A.=	.494	.341
R.P.M.=	206.27	206.27
I.H.P.=	179.00	177.64
S=	6.235	6.235
Blades=	3.	3.

Comparing the above results with those of Problem 5, it is evident that the failure to make the contract speed under the latter condition was due almost entirely to being limited to too small a diameter of propeller and to a too high number of revolutions, as under Problem 6, where the revolutions were the same as under Problem 5, but with the diameter increased to 6 feet 9 inches, the speed was easily made. Attention is called to the close agreement of chart I.H.P. and actual I.H.P. for Problem 6. If there is any error existing in the charts in this high range of block coefficients and of C, it results in giving probably slightly greater projected areas than are required for the work to be accomplished, though the actual screws for both conditions produced heavy vibrations, thus indicating lack of surface.

Problem 7. Choice of number of propellers and type of machinery as dictated by economy of propulsion.

Vessel having the following characteristics:

L.L.W.L.=416 feet; Greatest Immersed Beam=53 feet; Nominal B.C.=.63; $\frac{B}{L.W.L.}$ =.127; Draught=25 feet; Standard B.C.=.55 (Twin screw),=.675 (Single screw); Displacement = 10,018 tons; C = 1, (Twin screw) = 1.06 (Single screw); Appendage Resistance=.1335, (Twin screws)=.0295 (Single screw); E.H.P. (Bare Hull)=3,900; E.H.P. (All appendages) = 3900×1.1335 = 4421 (Twin screw),=3900×1.0295=4015 (Single screw); Speed=16 knots.

Machinery proposed:

Case 1.	*Case 2.*	*Case 3.*
Reciprocat'g engines.	Reciprocat'g engines.	Turbine engines.
Single screw.	Twin screws.	Twin screws.
Revolutions, 68.	Revolutions, 95.	Revs., 240.

Find I.H.P. for cases 1 and 2 and S.H.P. for case 3 and the propulsive coefficients that may be expected in each case, also characteristics of the propellers for each case.

	Blades.	Maximum tip speed.
Case 1. Max. Diameter of propeller = 22′–0″	4	4,712′
Case 2. Max. Diameter of propeller = 15′–6″	3	4,626′
Case 3. Max. Diameter of propeller = 11′–6″	3	8,671′

Case 1. Chart 5. Four-bladed Propeller.

T.S.	4,712	4,712	4,712	4,712
$T.S._1$	4,712	4,850	4,950	5,050
A (log)	0.00000	9.98747	9.97860	9.96992
A^4 (log)	0.00000	9.94988	9.91440	9.87968
A^8 (log)	0.00000	9.89976	9.82880	9.75936
$P.A._1 \div D.A._1$	.219	.225	.228	.2325
P.A. ÷ D.A.	.2919	.3366	.3701	.4089
$P.T._p$	7.03	7.16	7.22	7.32
$E.T._p$	7.85	7.98	8.10	8.22
$1 - s_1$	.8956	.8972	.8914	.8905
s_1	.1044	.1028	.1086	.1095
s	.1044	.1295	.1611	.1906
$1 - s$	.8956	.8705	.8389	.8094
S	16	16	16	16
S × 101.33 (log)	3.20986	3.20986	3.20985	3.20986
P × R (log)	3.25775	3.27009	3.28615	3.30170
S_1 (log)	1.20412	1.21724	1.23048	1.24559
E.H.P.	4,015	4,015	4,015	4,015
$E.H.P._1$ (log)	3.60369	3.64305	3.68277	3.72810
C	1.14	1.14	1.14	1.14
$P.C._1$	.6970	.6965	.6950	.6955
I.H.P.	6,567	7,195	7,890	8,764
P.C.	.611	.558	.5094	.458
D (feet)	23.21	22.1	21.56	21.09
P (feet)	28.01	27.44	27.78	28.16
R.p.m.	64.62	67.88	69.56	71.13

The desired propeller will have the following characteristics: D = 22 feet; P = 27.4 feet; R = 68; P.A. ÷ D.A. = .34; P.C. = .554; I.H.P. = 7,250.

Case 2. Chart 6. Twin Screws. Reciprocating Engines.

T.S.	4,626	4,622	4,626
T.S.$_1$	4,626	4,700	4,800
A (log)	0.00000	9.99311	9.98397
A^4 (log)	0.00000	9.97244	9.93588
A^8 (log)	0.00000	9.94488	9.87176
P.A.$_1$ ÷ D.A.$_1$	.253	.256	.261
P.A. ÷ D.A.	.253	.2728	.3025
P.T.$_p$	5.61	5.63	5.65
E.T.$_p$	6.10	6.12	6.15
$1 - s_1$	.9197	.9191	.9186
s_1	.0803	.0809	.0814
s	.0803	.09185	.1044
$1 - s$	.9197	.90815	.8906
S	16	16	16
S × 101.33 (log)	3.20986	3.20986	3.20986
P × R (log)	3.24621	3.25170	3.26018
S_1 (log)	1.20412	1.20932	1.21757
E H.P. (one screw)	2,211	2,211	2,211
E.H.P.$_1$ (log)	3.34459	3.36019	3.38494
C	1	1	1
P.C.$_1$	.699	.6985	.698
I.H.P.	6,326	6,562	6,952
P.C.	.699	.6737	.636
D (feet)	16.06	15.63	15.08
P (feet)	19.23	18.95	18.65
R.p.m.	91.7	94.23	97.61

The characteristics of the propeller will be: D = 15 feet 6 inches; P = 18 feet 10½ inches; R = 95; P.C. = .6658; I.H.P. = 6,640; P.A. ÷ D.A. = .292.

Case 3. Turbine Engines. Twin Screws. Chart 5.

T.S.	8,671	8,400	8,200	8,000	7,700
P.A. ÷ D.A.	.411	.391	.381	.371	.352
P.T.$_p$	9.36	9.24	9.17	9.10	8.96
E.T.$_p$	11.11	10.97	10.89	10.81	10.62
$1 - s$	.8425	.8423	.8420	.8418	.8437
s	.1575	.1577	.1580	.1582	.1563
S	16	16	16	16	16
S × 101.33 (log)	3.20986	3.20986	3.20986	3.20986	3.20986
P × R (log)	3.28429	3.28439	3.28455	3.28465	3.28367
S_1 (log)	1.20412	1.20412	1.20412	1.20412	1.20412
E.H.P., 1 screw	2,211	2,211	2,211	2,211	2,221
E.H.P.$_1$ (log)	3.34459	3.34459	3.34459	3.34459	3.34459
C	1	1	1	1	1
P.C.$_1$	.612	.625	.631	.639	.65
I.H.P.	7,224	7,071	7,007	6,919	6,802
S.H.P.	6,646	6,505	6,447	6,366	6,258
D (feet)	9.336	9.633	9.794	9.962	10.32
P (feet)	6.508	6.934	7.225	7.534	8.094
R.p.m.	295.6	277.6	267.5	255.6	237.4

Propellers.

	Case 1.	Case 2.	Case 3.
D = ...	22′ 00″	15′ 06″	10′ 02⅝″
P = ...	27′ 05″	18′ 10½″	7′ 10″
P.A. ÷ D.A. = . .	.34	.292	.356
I.H.P. = ...	7,250	6,640	6,825
P.C. = ...	.554	.6658	.648
R = ...	68	95	240

Considering the efficiency of propulsion alone, the single-screw vessel would be the most extravagant to drive, which is contrary to the generally accepted idea. This inefficiency of the single screw is due entirely to the diameter being too restricted for the low number of revolutions. Should we desire to drive the vessel at the same cost in I.H.P. as for the twin-screw vessel the Propulsive Coefficient required would equal 4,015 ÷ 6,640 = .6047; $P.C._1 = 1.14 \times .6047 = .6894$, and E.H.P. = 4,015 × 1.14 = 4,577.

For full diameter this would correspond (Chart 6) to a P.A. ÷ D.A. = .282; $P.T._p = 8.18$; $E.T._p = 9.3$; Tip Speed = 6,220.

The screw corresponding to these conditions would be as follows:

D = 18′ 09″;
P = 17′ 05″;
R = 105.66;
P.A. ÷ D.A. = .376.

A decrease in revolutions will be followed by an increase in diameter and in pitch, a decrease in P.A. ÷ D.A. and an increase in propulsive efficiency until the limiting diameter of 22 feet has been reached. To further decrease the revolutions below those corresponding to a full diameter of 22 feet, results in a problem of reduced diameter and a rapid falling off in propulsive efficiency.

Estimated Curves of Performance for Propellers of Reduced Diameter.

In such cases there is not only a reduction of diameter and of tip speed below that given by the Charts, but the Propulsive Coefficient actually realized is considerably less than that given by the Charts, the reduction in Propulsive Coefficient depending directly upon the amount of diameter decrease and the thrust deduction factor C.

If we turn to Problem 7, cases 1 and 2, we will find a value P.C. and a value $P.C._1$, given for Propulsive Coefficients, where P.C. is the actual propulsive coefficient obtained and $P.C._1$ is the Corresponding Chart Propulsive Coefficient. The ratio between P.C. and $P.C._1$, that is

$$P.C. \div P.C._1 = e = \frac{(A^8 - s_1)^3}{C \times [A^8 (1 - s_1)]^3}*$$

will be considered as applying to the designed propeller for all values of $I.T._d$ on Chart 3, so that to find the proper propulsive coefficient for a propeller of reduced diameter for any value of $I.T._d$, and for any value of P.A. ÷ D.A., the following formulas are used:

$$I.H.P. = \frac{D^2 \times P \times R \times I.T._d}{291.8 \times C^2 \times A^4} = \frac{P \times D \times T.S._1 \times A \times I.T._d}{291.8 \times \pi \times C^2 \times A^4}$$

$$= \frac{P \times D \times T.S._1 \times I.T._d}{916.7 \times C^2 \times A^3};$$

$$R = \frac{T.S._1 \times A}{\pi D} \text{ where } T.S._1 = \text{Tip Speed for } I.T._d \text{ from Chart 4;}$$

$$P.C. = P.C._1 \times e;$$

$$E.H.P. = I.H.P. \times P.C._1 \times e;$$

Speed = Speed from E.H.P. Curve corresponding to E.H.P.

$P.A._1 \div D.A._1$ (to enter Charts 3 and 4)

$= (P.A. \div D.A.) \times A^4$ for 3-bladed;

$= (P.A. \div D.A.) \times A^4 \times \frac{3}{4}$ for 4-bladed wheels.

* This reduction factor for Propulsive Coefficient consists of two factors; $\frac{1}{C}$ = factor due to thrust deduction and $\left\{\frac{A^8 - s_1}{A^8 (1 - s_1)}\right\}^3$ = factor due to limited diameter.

Differences in Characteristics of Propellers Designed from Charts 5 and 6.

To show the differences produced by designing propellers for the same conditions by Chart 5 and Chart 6, two problems will be solved and the resultant curves plotted on Fig. 3. This will illustrate how the diameter, projected area, and propulsive coefficient decrease and the slip increases for the same power and revolutions as we pass from Chart 6 condition to Chart 5 condition.

Suppose a vessel of Standard Block Coefficient of .5, Speed of 20 knots, E.H.P. (all appendages) applied to one screw = 1,000, the vessel being twin screw. Find series of propellers, revolutions varying, by Charts 6 and 5, for these conditions, also I.H.P. required. Full Chart Diameters can be carried.

The claim is often made that a large variation in projected area can be made without seriously affecting the efficiency of propulsion. That this claim is correct can readily be seen by consulting the Table of Comparison on Fig. 3, where are given columns of equal revolutions with corresponding pitches, diameters, projected-area ratios, and propulsive coefficients as obtained by Charts 5 and 6. It will be noted that Chart 6 gives for equal revolutions with Chart 5, slightly larger diameters, slightly lower pitches, much larger projected-area ratios, and practically equal propulsive cofficients. Suppose we take a compromise propeller between two propellers of equal diameter designed by the two Charts.

These propellers and the Compromise will be as follows:

	Chart 6.	Compromise.	Chart 5.
D=	8′ 00″	8′ 00″	8′ 00″
P=	9.01′	9.55′	9.9′
P.A.÷D.A.=	.340	.296	.252
P.C.=	.687	.689	.692
R=	250.	236.	223.

Should we decrease the projected-area ratio of the compromise screw, retaining the original pitch, the propeller will speed up slightly, depending upon the amount of decrease,

P.A.÷D.A.	T.S.	P.T.-p.	E.T.-p.	1—s.	E.H.P.	P.C.	2×I.H.P.	S.	101.33 S.	P×R (log).	D (feet).	P (feet).	R.P.M.
					BY CHART 6.								
.20	3,375	5.16	5.76	.8958	1,000	.7025	2,847	20	2,026.6	3.35453	11.18	23.54	96.09
.25	4,550	5.60	6.26	.8945	1,000	.6995	2,859	20	2,026.6	3.35516	9.593	15.01	151.
.30	5,580	5.79	6.50	.8908	1,000	.6935	2,884	20	2,026.6	3.35696	8.594	11.01	206.7
.35	6,550	5.81	6.60	.8802	1,000	.6845	2,922	20	2,026.6	3.36216	7.895	8.719	264.1
.40	7,130	5.75	6.56	.8765	1,000	.6720	2,976	20	2,026.6	3.36399	7.414	7.552	306.1
.45	7,800	5.61	6.41	.8752	1,000	.6560	3,049	20	2,026.6	3.36463	7.066	6.589	351.4
					BY CHART 5.								
.20	4,175	6.55	7.61	.8620	1,000	.698	2,865	20	2,026.6	3.37123	9.727	17.21	136.6
.25	5,480	7.67	8.97	.8551	1,000	.692	2,892	20	2,026.6	3.37472	8.013	10.89	217.7
.30	6,650	8.45	9.98	.8467	1,000	.677	2,954	20	2,026.6	3.37901	6.935	7.842	305.2
.35	7,650	8.95	10.69	.8372	1,000	.650	3,077	20	2,026.6	3.38391	6.204	6.167	392.5
.40	8,550	9.30	11.11	.8371	1,000	.619	3,231	20	2,026.6	3.38396	5.693	5.063	478.1
.45	9,500	9.55	11.47	.8326	1,000	.588	3,401	20	2,026.6	3.38630	5.276	4.247	573.2
.50	10,550	9.77	11.88	.8224	1,000	.558	3,584	20	2,026.6	3.39166	4.925	3.614	681.9
.55	11,780	9.95	12.38	.8037	1,000	.532	3,759	20	2,026.6	3.40165	4.599	3.092	815.4
.60	13,700	10.10	13.00	.7770	1,000	.506	3,953	20	2,026.6	3.41632	4.297	2.57	1,015.0
.625	15,300	10.17	13.36	.7612	1,000	.494	4,049	20	2,026.6	3.42524	4.153	2.27	1,173.0

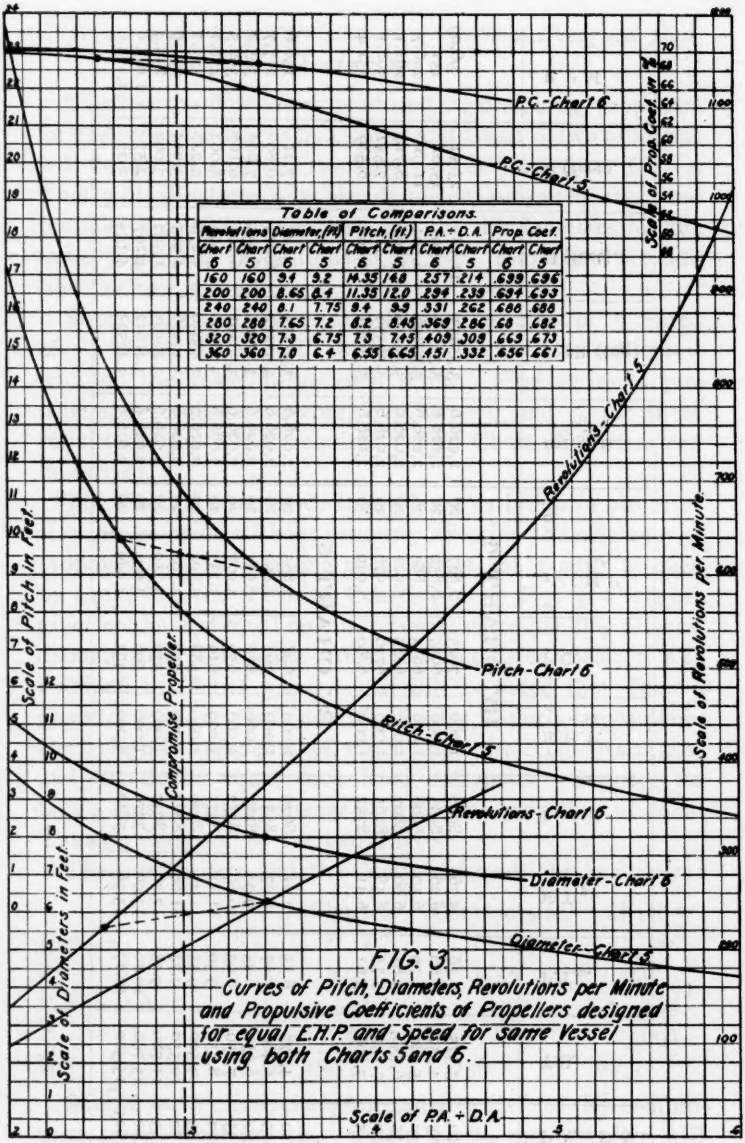

P.C. - Chart 6
P.C. - Chart 5
Scale of Prop. Coef. in %
Revolutions - Chart 5
Scale of Revolutions per Minute
Pitch - Chart 6
Pitch - Chart 5
Revolutions - Chart 6
Diameter - Chart 6
Diameter - Chart 5
Compromise Propeller.
Scale of Pitch in Feet.
Scale of Diameters in Feet.
Scale of P.A. ÷ D.A.

Table of Comparisons.

Revolutions		Diameter, (ft)		Pitch, (ft)		P.A. ÷ D.A.		Prop. Coef.	
Chart 6	Chart 5	Chart 6	Chart 5	Chart 6	Chart 5	Chart 6	Chart 5	Chart 6	Chart 5
160	160	9.1	9.2	14.35	14.8	.257	.214	.699	.696
200	200	8.65	8.4	11.35	12.0	.294	.239	.694	.693
240	240	8.1	7.75	9.4	9.9	.331	.262	.686	.686
280	280	7.65	7.2	8.2	8.45	.369	.286	.68	.682
320	320	7.3	6.75	7.3	7.95	.409	.309	.663	.673
360	360	7.0	6.4	6.55	6.65	.451	.332	.636	.661

FIG. 3.

Curves of Pitch, Diameters, Revolutions per Minute and Propulsive Coefficients of Propellers designed for equal E.H.P. and Speed for same Vessel using both Charts 5 and 6.

FIG. 3.

and the slip will increase, while if the projected area be increased, the effect will be the opposite. In either case, the propulsive efficiency will probably not change more than the extreme limits given by the screws for Charts 5 and 6.

Tug-Boat Propellers.

Professor Peabody, in writing on this subject, states as follows:

"A towboat, especially of the harbor type, has three various kinds of duty, namely: (1) To run free, passing from place to place, and perhaps seeking business in face of competition; (2) towing at a reduced speed, commonly at half speed or somewhat more, and (3) pulling or pushing a ship into her berth at a very slow speed, or nearly without speed."

For such duties, the propeller adopted must be a compromise and must be designed to do its duty of minor importance without excessive sacrifice of its efficiency when performing the major important duty. Thus, a Navy sea-going tug boat must be able to steam economically with the fleet at the maximum cruising speed of the fleet, and at the same time must be fairly efficient when towing a vessel of moderate resistance.

Suppose the estimated maximum continuous speed of the tug boat is 14 knots, the I.H.P. of the engine, single screw, being 1,800 and the revolutions for this power 120. The maximum diameter of the screw that can be fitted is 12 feet 9 inches. The propeller is required to work with maximum efficiency when its resistance has been increased by a tow to such an extent that the maximum speed when developing full power at the designed revolutions is 13 knots. The screw to be 4-bladed.

The dimensions of the vessel are as follows: L.L.W.L. = 175; B ÷ L.W.L. = .195; Greatest Immersed Beam = 34 feet 1½ inches; Nominal B.C. = .531; Standard B.C. (Line Z) = .74; Draught (mean) = 12 feet 6 inches; Displacement

= 1,135 tons; Coefficient of Immersed Midship Section = .875; C for Standard B.C. = 1.27; Maximum diameter = 12 feet 9 inches; R.p.m. = 120; Maximum T.S. = 4,807; I.H.P. = 1,800.

Chart.	6	5	6	5	6	5
T.S.	4,807	4,807	4,807	4,807	4,807	4,807
$T.S._1$	4,807	4,807	4,950	4,950	5,050	5,050
A (log)	0.00000	0.00000	9.98726	9.98726	9.97858	9.97858
A^4 (log)	0.00000	0.00000	9.94904	9.94904	9.91432	9.91432
A^8 (log)	0.00000	0.00000	9.89808	9.89808	9.82864	9.82864
$P.A._1 \div D.A._1$	.261	.223	.269	.229	.273	.233
P.A. ÷ D.A.	.261	.223	.3025	.2575	.3325	.2838
P.T.	5.66	7.11	5.7	7.25	5.71	7.32
E.T.	6.02	7.88	6.08	8.00	6.10	8.11
$1 - s_1$	.9402	.9023	.9375	.9031	.9362	.9038
s_1	.0598	.0977	.0625	.0969	.0638	.0962
s	.0598	.0977	.07903	.1225	.09466	.1427
$1 - s$	.9402	.9023	.92097	.8775	.90534	.8573
S	13	13	13	13	13	13
S × 101.33 (log)	3.11968	3.11968	3.11968	3.11968	3.11968	3.11968
P × R (log)	3.14646	3.16433	3.15543	3.17643	3.16287	3.18655
S_1 (log)	1.11394	1.11394	1.12166	1.12643	1.12850	1.13688
I.H.P.	1,800	1,800	1,800	1,800	1,800	1,800
C	1.27	1.27	1.27	1.27	1.27	1.27
$I.T._d$	2.13	2.26	2.21	2.38	2.27	2.45
D (feet)	15.71	14.9	14.36	13.51	13.50	12.64
P (feet)	14.35	14.25	13.42	13.25	12.83	12.70
R.p.m.	97.65	102.7	106.6	113.3	113.4	121
$P.A._{II} \div D.A._{II}$	.381	.297	.4032	.3433	.4433	.3784
$P.C._1$ (charts)	.6985	.6965	.6976	.696	.6974	.695
e (log)	9.89620	9.89620	9.87280	9.85843	9.85258	9.82741
$P.C. = P.C._1 \times e$	.55	.5484	.5205	.5024	.4967	.4671

The propellers obtained will be as follows, the curves for Chart 6 propeller having been extended to a diameter of 12 feet 9 inches:

	Chart 6.	Chart 5.
Diameter,	12′ 09″	12′ 09′
Pitch,	12′ 06½″	12′ 08′
P.A. ÷ D.A.,	.491	.374
R.p.m.,	120	120
I.H.P.,	2,286	2,286
Towing speed,	13	13
P.C.,	.480	.472

Should it be estimated that the engine has considerable reserve power the propeller from Chart 6 should be used; but if the engine is built with little reserve power the Chart

5 propeller should be adopted, as it would be considerably lighter than the other.

Methods of Estimating Power.

These are four in number, as follows:

1. The Admiralty Coefficient.
2. The Law of Comparison.
3. Independent Estimate.
4. Model Experiments.

The usual methods of estimating power, employed in the Bureau of Steam Engineering, are Nos. 1, 2 and 4.

Admiralty Coefficient.

The equation in which this coefficient occurs is

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}}V^3}{K},$$

in which

I.H.P. = Indicated horsepower of the engine;
D = Displacement, in tons.
V = Speed, in knots per hour.
K = Admiralty Coefficient.

Where the Shaft Horsepower is given instead of the Indicated Horsepower, the equation may read:

$$\text{S.H.P.} = \frac{D^{\frac{2}{3}}V^3 \times .92}{K}.$$

This coefficient must be derived from some ship for which the displacement, power, and speed are known, and further, in order that a close agreement may be expected between the estimated speed and the actual trial speed of the ship, the coefficient must be derived from a ship that is geometrically similar to the ship under design, and which has the corresponding speed. These terms will be explained when the "Law of Comparison" is taken up. Furthermore, it is absolutely necessary that the conditions existing in the new

ship are such as will permit the realization of an equal propulsive coefficient of propulsion with the compared vessel. Where these conditions exist, we may write

$$\text{E.H.P.} = \frac{D^{\frac{1}{3}} V^3 \times \text{P.C.}}{K}.$$

A moderate deviation in the first two requirements may not seriously affect the value of the method, but such is not the case with deviation from the third requirement.

Law of Comparison.

1. "Corresponding Speeds."—The corresponding speeds for similar ships are proportional to the square roots of their lengths.

2. "Displacements."—Similar ships have displacements proportional to the cubes of their lengths.

3. "Corresponding Speeds."—The corresponding speeds for similar ships are proportional to the sixth roots of their displacements.

4. "Horsepowers."—The horsepowers of similar ships at corresponding speeds are proportional to the seven-sixths powers of their displacements.

This rule (4), is not strictly correct, however, as the frictional resistance does not follow the law of mechanical similitude.

5. "Variation of Power with Speed."—Where the difference between the two speeds compared is small, we may assume that "The power for a ship is proportional to the cube of the speed," although this exponent may be widely departed from at high speeds.

6. "Variation of Power with Variation in Displacement."—For small changes in draught, we may assume that "The powers vary as the nth power of the Displacements," where n may vary from 2/3 for large ships of moderate speed to 1/6 for ships and boats of high speed.

In comparisons of hulls for similarity of form, Chart 1

should always be employed on account of the great influence of $\frac{B}{L.L.W.L.}$ and coefficient of immersed midship section on fullness of lines.

Independent Estimate.

The tow-rope resistance of a vessel is divided into two parts, surface or frictional resistance and residual resistance. The residual resistance is again divided into wave-making, eddy-making, and stream-line resistance.

The equation used for the calculation of frictional resistance is

$$R_f = fSV^n,$$

in which R_f is the force, in pounds, required to overcome the surface resistance, S is the wetted surface, in square feet, and V is the speed, in knots per hour. f and n are quantities taken from tables which can be obtained from any work on the "Resistance of Ships," and which are included here.

The equation used for finding the residual resistance is given as

$$R_w = \frac{bD^{\frac{1}{3}}V^4}{L},$$

where D, V and L are the displacement, in tons, the speed in knots per hour, and the length on the load water line, in feet. b is a numerical factor, having a value for long, fine ships of about .35; moderately fine ships, .40; ships broad in proportion to length but with fine ends, .45; freighters, .5. The value of b is also likely to be affected by speed, especially when the speed-length ratio is high.

Total Resistance.—As stated before, this is the sum of the two resistances, frictional and residual, and the equation for it is

$$R = R_f + R_w = fSV^n + \frac{bD^{\frac{1}{3}}V^4}{L}.$$

Using this equation in the estimation of the E.H.P., the equation for net E.H.P. takes the form

$$\text{E.H.P.} = 0.00307\left(fSV^{n+1} + \frac{bD^{\frac{1}{3}}V^5}{L}\right),$$

where the various letters have the same significance as before.

Wetted Surface.—This is determined from the lines of the ship and is a tedious operation. The surface is computed in square feet. For a preliminary design, the wetted surface may be computed by the equation $S = C_V \sqrt{DL}$, where D is the displacement, in tons, L the length on load-water line, and C a coefficient depending on the beam and draught.

Model Experiments.—The fourth method for determining power is by aid of model experiments in a towing basin. To illustrate the method, suppose that the tow-rope resistance for a paraffin model 20 feet long is 12.8 pounds, when towed at the speed corresponding to 25 knots for the full-sized vessel which has a length on load-water line of 700 feet, then

$$V_m : 25 :: 20 : 700 \quad \therefore \quad V_m = 4.23 \text{ knots.}$$

The wetted surface of the vessel is 67,540 square feet, therefore the wetted surface of the model:

$$S_m : 67{,}540 :: 20^2 : 700^2 \quad \therefore \quad S_m = 55.1 \text{ square feet.}$$

The friction factor and the exponent taken from Froude's tables are

$$f = 0.00834 \text{ and } n = 1.94;$$

therefore the frictional resistance is

$$0.00834 \times 55.1 \times 4.23^{1.94} = 7.54 \text{ pounds.}$$

The total frictional resistance of the full-sized vessel, $f = 0.00847$ and $n = 1.825$, is $0.00847 \times 67{,}540 \times 25^{2.825}$ and the E.H.P. (frictional)

$$0.00307 \times 0.00847 \times 67{,}540 \times 25.^{2.825} = 15{,}600.$$

Taking the frictional resistance of the model from the total tow-rope resistance of the model, gives for the residual resistance

$$12.8 - 7.54 = 5.26 \text{ pounds.}$$

The corresponding residual resistance for the ship is

$$R_w : 5.26 :: 700^3 : 20^3 \quad \therefore \quad R_w = 225{,}500 \text{ pounds.}$$

At 25 knots the E.H.P. required to overcome this residual resistance will be $0.00307 \times 225{,}500 \times 25 = 17{,}310$.

The total E.H.P. will then be $15{,}600 + 17{,}310 = 32{,}910$.

In all the above methods, the results obtained are those for the bare hull only, and the appendage resistance increase called for by Chart 2 must be applied before we are in a position to correctly compute the propeller and the indicated and shaft horsepowers.

(Credit must be given Peabody's work on "Propellers" for the major part of the above sections on "Resistance of Ships." —C. W. D.)

Design of Propellers by Comparison.

Sometimes, in designing the propellers for a vessel, we may wish to obtain propellers which will give an equal propulsive efficiency with those fitted to an earlier vessel of similar form but of different size, and whose performance has been regarded as excellent.

To obtain such propellers, we can work directly with the Chart formulas, always remembering that, according to the Charts, for equal propulsive efficiencies the projected area ratios, thrusts (effective, propulsive and indicated), and tip speeds must remain equal. Bearing this in mind, and obtaining the ratios between the equations for diameter, pitch, and revolutions of the existing propellers and those of the propellers we are designing, the following equations result:

$$D_2 = D_1\sqrt{\frac{E.H.P._2 \times S_1}{E.H.P._1 \times S_2}} = D_1\sqrt{\frac{I.H.P._2 \times S_1}{I.H.P._1 \times S_2}} = D_1\sqrt{\frac{S.H.P._2 \times S_1}{S.H.P._1 \times S_2}}.$$

$$P_2 = P_1\sqrt{\frac{E.H.P._2 \times S_2}{E.H.P._1 \times S_1}} = P_1\sqrt{\frac{I.H.P._2 \times S_2}{I.H.P._1 \times S_1}} = P_1\sqrt{\frac{S.H.P._2 \times S_2}{S.H.P._1 \times S_1}}.$$

$$R_2 = R_1\sqrt{\frac{E.H.P._1 \times S_2}{E.H.P._2 \times S_1}} = R_1\sqrt{\frac{I.H.P._1 \times S_2}{I.H.P._2 \times S_1}} = R_1\sqrt{\frac{S.H.P._1 \times S_2}{S.H.P._2 \times S_1}}.$$

Where

D_1 = Diameter of existing propeller;

D_2 = Diameter of propeller for new ship;

P_1 = Pitch of existing propeller;

P_2 = Pitch of propeller for new ship;
R_1 = Revolutions of existing propeller;
R_2 = Revolutions of propeller for new ship;
S_1 = Speed of existing vessel;
S_2 = Speed of new ship;
E.H.P.$_1$ = Effective horsepower for S_1 of existing vessel;
I.H.P.$_1$ = Indicated horsepower for S_1 of existing vessel;
S.H.P.$_1$ = Shaft horsepower for S_1 of existing vessel;
E.H.P.$_2$ = Effective horsepower for S_2 of new vessel;
I.H.P.$_2$ = Indicated horsepower for S_2 of new vessel;
S.H.P.$_2$ = Shaft horsepower for S_2 of new vessel.

Should the speeds used in the above be the corresponding speeds by Froude's Law of Comparison, where

$$S_2 = S_1 \left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{1}{6}} \text{ and}$$

$$\text{Power}_2 = \text{H.P.}_2 = \text{H.P.}_1 \left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{7}{6}},$$

the following forms will obtain:

$$D_2 = D_1\left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{1}{3}} = D_1\left(\frac{L_2}{L_1}\right)^{\frac{3}{2}} = D_1\left(\frac{S_2}{S_1}\right)^3 = D_1\sqrt{\frac{\text{H.P.}_2 \times S_1}{\text{H.P.}_1 \times S_2}};$$

$$P_2 = P_1\left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{2}{3}} = P_1\left(\frac{L_2}{L_1}\right)^2 = P_1\left(\frac{S_2}{S_1}\right)^4 = P_1\sqrt{\frac{\text{H.P.}_2 \times S_2}{\text{H.P.}_1 \times S_1}};$$

$$R_2 = R_1\left(\frac{\text{Displacement}_1}{\text{Displacement}_2}\right)^{\frac{1}{3}} = R_1\left(\frac{L_1}{L_2}\right)^{\frac{3}{2}} = R_1\left(\frac{S_1}{S_2}\right)^3 = R_1\sqrt{\frac{\text{H.P.}_1 \times S_2}{\text{H.P.}_2 \times S_1}}.$$

According to these formulas it appears that the "Law of Mechanical Similitude" does not apply to screw propellers, as the diameters are seen to vary approximately as the cubes of the speeds, while the pitches vary, with the same degree of approximation, as the fourth power.

CAVITATION.

Taylor, in his work on "Speed and Power of Ships," states as follows concerning this phenomenon:

"Nature of Cavitation.—The phenomenon known as cavi-

tation has been given much attention, of late years, in connection with quick-running turbine-driven propellers. It appears to have been first identified upon the trials, in 1894, of the torpedo-boat destroyer *Daring*, which had reciprocating engines. When driven at full power with the original screws this vessel showed very serious vibration, evidently due to some irregular screw action. The propulsive efficiency was poor, the maximum speed obtained being 24 knots for 3,700 I.H.P. and 384 revolutions per minute.

"Mr. Sidney W. Barnaby, the engineer of the Thornycrofts, who built the *Daring*, came to the conclusion that at high thrust per square inch at which the screws were working the water was unable to follow up the screw blades, and that 'the bad performance of the screws was due to the formation of cavities in the water forward of the screw, which cavities would probably be filled with air and water vapor.' So Mr. Barnaby gave the phenomenon the name of cavitation. The screws which gave the poor results had a diameter of 6 feet 2 inches, a pitch of 8 feet $7\frac{3}{4}$ inches and a blade area of 8.9 square feet. Various alternative screws were tried, and the trouble was cured by the use of screws 6 feet 2 inches in diameter, 8 feet 11 inches pitch and 12.9 square feet blade area. With these screws 24 knots was obtained with 3,050 I.H.P., and the maximum speed rose from 24 knots to over 29 knots."

For the *Daring* cavitation appeared to begin when the screw area was such that the thrust per square inch of projected area was a little over 11 pounds per square inch; $11\frac{1}{4}$ pounds is the figure given by Mr. Barnaby. "For a time it was thought that the thrust per square inch of projected area was a satisfactory criterion in connection with cavitation, and that the limiting thrust per square inch of projected area found on the *Daring* was generally applicable."

"This, however, is not the case. Greater thrusts have been successfully used and cavitation is liable to appear at much lower thrusts. In one case, within the author's experience, cavitation appeared when the thrust was about 5

pounds per square inch of projected area, the tip speed being about 5,000 feet per minute, and in another when the thrust was about 7.5 pounds and the tip speed about 6,500 feet per minute. There is little doubt that the prime factors involved in cavitation are: (1) The speed of the blade through the water, which is evidently measured by the tip speed, and (2) the shape of the blade section."

While Mr. Taylor may be correct in his statements as to the prime factors involved in cavitation, the writer is of the opinion that as all of the elements of the propeller, namely, pitch, diameter, projected area and revolutions, and in addition, the form of the after body of the vessel behind which the propeller is working, each has its influence, it would be difficult, if not impossible, to differentiate between them as to to their relative effect. Personally, he is of the opinion that the prime factor in regulating the thrusts that can safely be used on any particular propeller, is the form of the after body of the hull. As the after body fines, the thrust may be increased, and *vice versa*. Thus Chart 5 is derived from the actual performance of numerous vessels, and the values of $E.T._p$ given for the different standard block coefficients are considered as those which can safely be used without noticeable cavitation occurring. It is estimated that we may exceed these Chart 5 thrusts by at least 10 per cent. without the vibration from cavitation becoming excessive.

The thrusts given on Chart 5 are unaffected by either thrust deduction or reduction of diameter. Considering these two latter elements, there result three formulas for obtaining effective thrust, viz:

(1) Net Effective Thrust

$$= E.T._p = \frac{2.88 \times E.H.P.}{(P.A. \div D.A.) \times S \times D^2}.$$ Full diameter and no thrust deduction.

(2) Net Effective Thrust

$$= \frac{E.T._p}{C^3} = \frac{2.88 \times E.H.P.}{(P.A. \div D.A.) \times S \times D^2}.$$ Full diameter and thrust deduction.

(3) Net Effective Thrust

$$=\frac{E.T._p}{C^3}\times\left\{\frac{A^8-s_1}{A^5(1-s_1)}\right\}^2=\frac{2.88\times E.H.P.}{(P.A.\div D.A.)\times S\times D^2}.$$ Reduced diameter and thrust deduction.

Thus, by (1), the E.T.$_p$'s obtained would be those which are shown on Chart 5, while by (2), they would be reduced by the cube of the thrust deduction factor, and by (3), in addition to the reduction due to thrust deduction, the diameter reduction factor also enters.

Let us take a value of P.A.÷D.A.=.3 and also=.6 on Chart 5 and tabulate the E.T.$_p$'s for the Block Coefficients from .9 to .2, both with and without the thrust deduction factor, and allow an overload of 10 per cent., the following results:

Standard Block Coefficients.

P.A.÷D.A.	...	.2	.3	.4	.5	.6	.7	.8	.9
.3	(1)*	10.62	10.41	10.19	9.98	9.77	9.57	9.33	9.02
	10 p.c.	11.68	11.45	11.20	10.98	10.75	10.53	10.23	9.92
	(2)*	10.62	10.41	10.19	9.98	8.73	5.70	3.43	2.20
	10 p.c.	11.68	11.45	11.20	10.98	9.60	6.27	3.77	2.42
	T.S.	6,650	6,650	6,650	6,650	6,650	6,650	6,650	6,650
.6	(1)*	13.56	13.39	13.21	13.00	12.77	12.45	12.01	11.55
	10 p.c.	14.92	14.72	14.53	14.30	14.05	13.70	13.21	12.71
	(2)*	13.56	13.39	13.21	13.00	11.40	7.41	4.42	2.81
	10 p.c.	14.92	14.72	14.53	14.30	12.54	8.15	4.86	3.09
	T.S.	13,600	13,600	13,600	13,600	13,600	13,600	13,600	13,600

* (1) Without thrust deduction. (2) With thrust deduction.

From this table, based on actual performances, it is seen that with a P.A.÷D.A. of .3 and a constant tip speed of 6,650 feet per minute, the effective thrusts for safe design, taking into account the thrust deduction, vary from 2.2 pounds per square inch of projected area, the vessel having standard B.C. of .9, to 10.62 pounds with a standard B.C. of .2, and that these values of thrust can be increased to 2.42 and 11.68 pounds respectively without the vibration becoming excessive.

Table of C, for Wetted Surface.

B÷H	C	B÷H	C	B÷H	C
2.0	15.63	2.5	15.50	3.0	15.62
2.1	15.58	2.6	15.51	3.1	15.66
2.2	15.54	2.7	15.53	3.2	15.71
2.3	15.51	2.8	15.55	3.3	15.77
2.4	15.50	2.9	15.58	3.4	15.83

B=beam. H=draught.

FROUDE'S SURFACE-FRICTION CONSTANTS.

Given by Taylor.

Surface-Friction Constants for Paraffin Models in Fresh Water. Exponent n=1.94.

Length, feet.	Coefficient.	Length, feet.	Coefficient.	Length, feet.	Coefficient.
2.0	0.01176	10.0	0.00937	14.0	0.00883
3.0	0.01123	10.5	0.00928	14.5	0.00877
4.0	0.01083	11.0	0.00920	15.0	0.00873
5.0	0.01050	11.5	0.00914	16.0	0.00864
6.0	0.01022	12.0	0.00908	17.0	0.00855
7.0	0.00997	12.5	0.00901	18.0	0.00847
8.0	0.00973	13.0	0.00895	19.0	0.00840
9.0	0.00953	13.5	0.00889	20.0	0.00834

Surface-Friction Constants for Painted Ships in Sea Water. Exponent n = 1.825.

Length, feet.	Coefficient.	Length, feet.	Coefficient.	Length, feet.	Coefficient.
8	0.01197	40	0.00981	180	0.00904
9	0.01177	45	0.00971	200	0.00902
10	0.01161	50	0.00963	250	0.00897
12	0.01131	60	0.00950	300	0 00892
14	0.01106	70	0.00940	350	0.00889
16	0.01086	80	0.00933	400	0.00886
18	0.01069	90	0.00928	450	0.00883
20	0.01055	100	0.00923	500	0.00880
25	0.01029	120	0.00916	550	0.00877
30	0.01010	140	0.00911	600	0.00874
35	0.00993	160	0.00907		

Surface-Friction Constants. Exponent, 1.825.

Given by Denny.

Length, feet.	Coefficient.	Length, feet.	Coefficient.	Length, feet.	Coefficient.
40	0.00996	260	0.00870	550	0.00853
60	0.00957	280	0.00868	600	0.00850
80	0.00933	300	0.00866	650	0.00848
100	0.00917	320	0.00864	700	0.00847
120	0.00905	340	0.00863	750	0.00846
140	0.00896	360	0.00862	800	0.00844
160	0.00889	380	0.00861	850	0.00842
180	0.00884	400	0.00860	900	0.00841
200	0.00879	420	0.00859	950	0.00840
220	0.00876	450	0.00858	1,000	0.00839
240	0.00872	500	0.00855		

TIDEMAN'S SURFACE-FRICTION CONSTANTS.

Derived from Froude's Experiments.

Surface-Friction Constants for Ships in Salt Water of 1.026 Density.

Length of ship in feet.	Iron bottom clean and well painted.		Copper or zinc sheathed.			
			Sheathing smooth and in good condition.		Sheathing rough and in bad condition.	
	f	*n*	*f*	*n*	*f*	*n*
10	0.01124	1.8530	0.01000	1.9175	0.01400	1.8700
20	0.01075	1.8490	0.00990	1.9000	0.01350	1.8610
30	0.01018	1.8440	0.00903	1.8650	0.01310	1.8530
40	0.00998	1.8397	0.00978	1.8400	0.01275	1.8470
50	0.00991	1.8357	0.00976	1.8300	0.01250	1.8430
100	0.00970	1.8290	0.00966	1.8270	0.01200	1.8430
150	0.00957	1.8290	0.00953	1.8270	0.01183	1.8430
200	0.00944	1.8290	0.00943	1.8270	0.01170	1.8430
250	0.00933	1.8290	0.00936	1.8270	0.01160	1.8430
300	0.00923	1.8290	0.00930	1.8270	0.01152	1.8430
350	0.00916	1.8290	0.00927	1.8270	0.01145	1.8430
400	0.00910	1.8290	0.00926	1.8270	0.01140	1.8430
450	0.00906	1.8290	0.00926	1.8270	0.01137	1.8430
500	0.00904	1.8290	0.00926	1.8270	0.01136	1.8430

Where the P.A.÷D.A. has been increased to .6, these safe thrusts become 2.81 and 13.56 pounds, and the overloads 3.09 and 14.92 pounds respectively, the tip speeds for the safe loads having a constant value of 13,600 feet per minute.

In many cases where vibration is charged to cavitation, it may be due to an entirely different cause, that is, too close proximity of the blade tips to the hull or too close proximity of the leading edge of the blades to the after edge of the struts or stern post. Such vibration is generally localized, however, and becomes unnoticeable when the observer moves from the locality to which the vibration is transmitted by the hull structure.

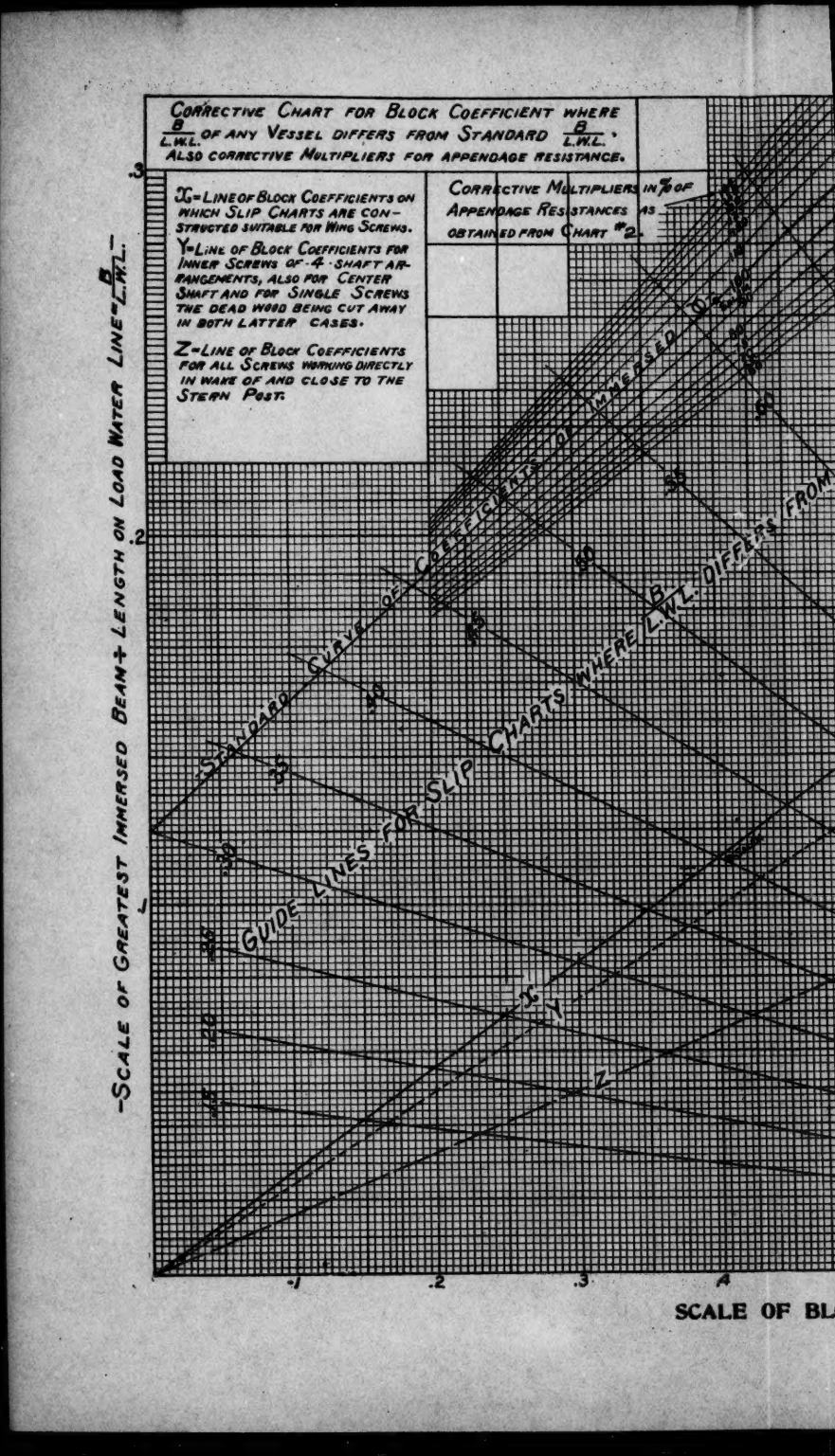
Symbols, definitions and formulas for propeller design.		Problems where Full data for design. Hull data. { "L.L.W.L." | "Block Coef." / "B." | "B" / "Displ." | L.W.L.. "Revs."—"Diam."—"E.H.P. (bare hull)." "S."—"Stand. B.C."—"C."—Form No.
T.S.=Tip speed allowed	1	log (πR.D.) (allowed R. & D.)=log (1)
T.S.$_1$=or>T.S.=Tip speeds f'm Charts 5 & 6.	2	log T.S.$_1$=log (2)
A.=T.S.÷T.S.$_1$	3	log (1)—log (2)=log (3)
A.^{4}	4	4 log (3)=log (4)
A.^{8}	5	8 log (3)=log (5)
P.A.$_1$÷D.A.$_1$=Projected area ratio from Charts 5 or 6 corresponding to T.S.$_1$	6	log (6)
P.A.÷D.A.=(P.A.$_1$÷D.A.$_1$)÷A.^{4}=Projected area ratio of final 3-bladed screw.	7	log (6)—log (4)=log (7)
I.T.$_4$=Indicated thrust, in pounds, from Charts 5 or 6 for T.S.$_1$	8	
P.T.$_p$=Propulsive thrusts for T.S.$_1$	9	log (9)
E.T.$_p$=Effective thrusts for T.S.$_1$	10	log (10)
1—s_1=P.T.$_p$÷E.T.$_p$	11	log (9)—log (10)=log (11)
$s_1=1-\frac{\text{P.T.}_p}{\text{E.T.}_p}$=Initial apparent slip	12	(12)
$s=\frac{s_1}{A^8}$=Actual apparent slip	13	log (12)—log (5)=log (13)
1—s	14	(14)
S=Designed speed of ship	15	log (15)

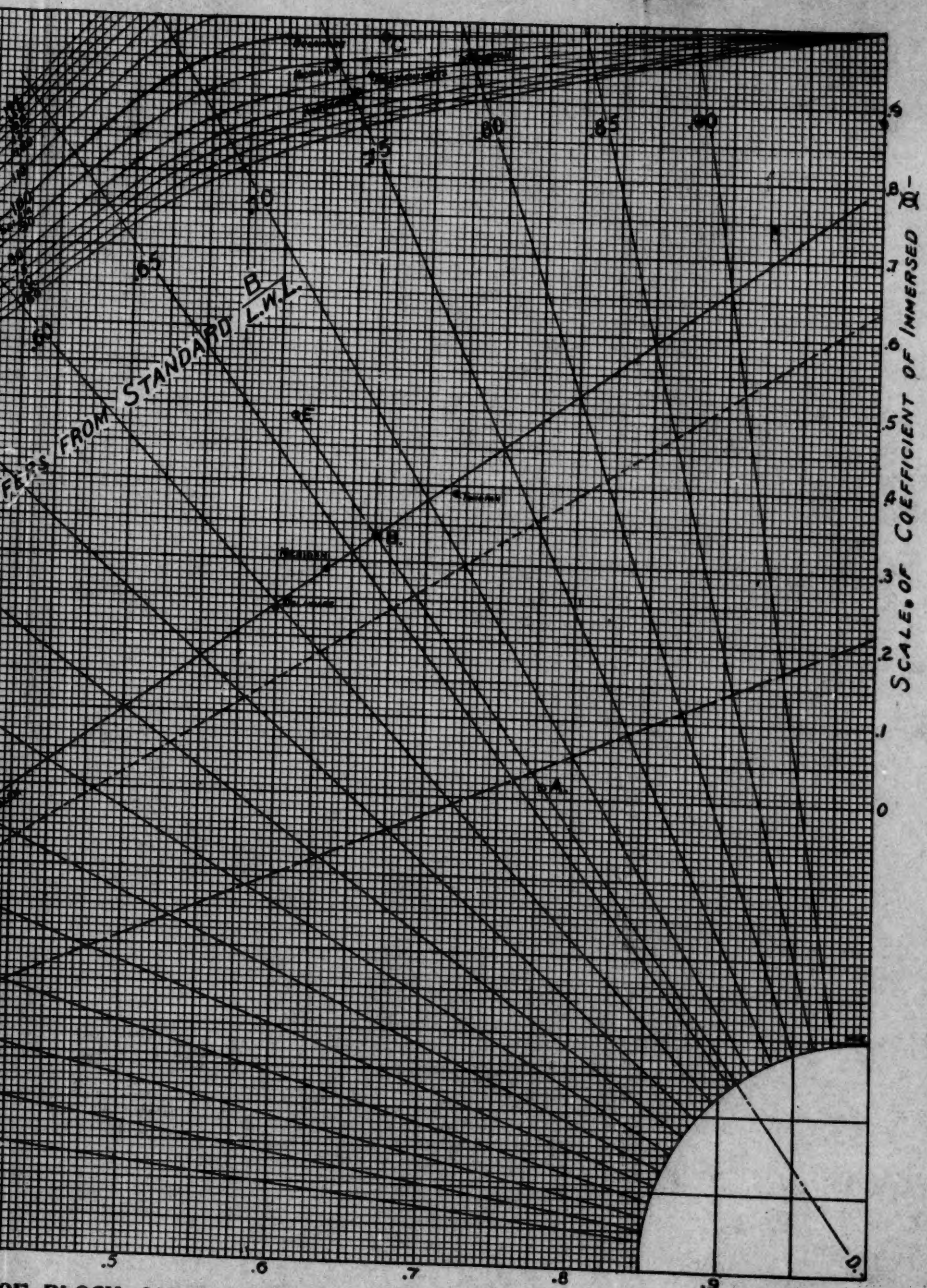
FOR COMPUTATION OF SCREW PROPELLERS.

Problems where diameter is limited.		Problems where diameter is unlimited.	
gn.	Insufficient data for design.	Full data for design.	Insufficient data for design.
Block Coef." "B" L.W.L.. H.P. (bare —Form No. 1.	Hull data. {"L.L.W.L." "B." "Displ." | "Block Coef." "B." L.W.L.} Revs."—"Diam."—"S."—"Estimated I.H.P." "Stand. B.C."—"C."—Form No. 2.	Same hull data as Form No. 1. Diameter and revolutions unlimited. Form No. 3.	Same hull data as Form No. 2. Diameter and revolutions unlimited. Form No. 4.
.)=log (1)...	log (1)	.....	
.....	log (2)	log (2)	log (2).
.....	log (3)	.....	
.....	log (4)	.....	
.....	log (5)	.....	
.....	log (6)	log (6)	log (6).
.....	log (7)	.....	
.....	log (8)	.....	log (8).
.....	log (9)	log (9)	log (9).
.....	log (10)	log (10).....	log (10).
.....	log (11)	log (11)	log (11).
.....	log (12)	(12)	(12).
.....	log (13)	(13)=(12).....	(13)=(12).
.....	log (14)	log (14)=log (11).....	log (14)=log (11).
.....	log (15)	log (15)	log (15).

Symbols, &c.		Form No. 1.	
101.33 S = Speed, feet per minute	16	2.00574 + log (15) = log (16)	log
$P. \times R. = \frac{101.33\ S}{1-s}$ = Pitch × revolutions	17	log (16) − log (14) = log (17)	log
$S._1 = \frac{P. \times R.\ (1-s_1)}{101.33}$ = Speed to use in diameter formulas.	18	log (17) + log (11) − 2.00574 = log (18)	.....
E.H.P. = Effective H.P. for S.....	19	log (19)	.....
$E.H.P._1 = E.H.P. \times \left[\frac{S_1}{S}\right]^3$ = E.H.P. for S_1	20	log (19) + 3 {log (18) − log (15)} = log (20)	.....
C = Thrust deduction factor from Chart 7.....	21	log (21)	log
$E.H.P._{.11} = C \times E.H.P._{.1}$	22	log (20) + log (21) = log (22)	.....
$P.C._1$ = Propulsive coefficient for $T.S_1$.....	23	log (23)	.....
$I.H.P. = \frac{E.H.P._{.11}}{P.C._{.1}}$	24	log (22) − log (23) = log (24)	log
$P.C. = \frac{E.H.P.}{I.H.P.} = \frac{P.C._1}{C.} \times \left[\frac{A^s - s_1}{A^s\ (1-s_1)}\right]^s$	25	log (19) − log (24) = log (25)	.....
$D = \text{Diam.}_{(3\ \text{blades})} = \sqrt{\frac{2.88 \times E.H.P._{.1} \times C.^s}{(P.A. \div D.A.) \times E.T._p \times S._1}}$	26	$\frac{1}{2}$ { 0.45939 + log (20) + 3 log (21) − log (7) + log (10) + log (18) = log (26) }	.....
$D = \text{Diam.}_{(3\ \text{blades.})} = \sqrt{\frac{291.8 \times I.H.P \times C^2}{P. \times R \times I.T._d}}$	27	.....	$\frac{1}{2}$ {
$D = \text{Diam.}_{(4\ \text{blades})} = \sqrt{\frac{2.88 \times E.H.P_1 \times C.^s \times .865}{(P.A. \div D.A.) \times E.T._p \times S._1}}$	28	For 4-bladed screws use log (28) instead of (26).	.....
$D = \text{Diam.}_{(4\ \text{blades})} = \sqrt{\frac{291.8 \times I.H.P. \times C^3 \times .865}{P. \times R. \times I.T._d.}}$	29	.....	Fo
$P. = \text{Pitch} = \frac{P. \times R. \times \pi D.}{T.S.}$	30	0.49715 + log (17) + log (26) − log (1) = log (30).	0.4
$R. = \text{Revolutions} = \frac{T.S.}{\pi D.}$	31	log (1) − (0.49715 + log (26)) = log (31)	log
$P.A._{.11} \div D.A._{.11}$ (4-bladed screw) $= \frac{P.A.}{D.A.} \times \frac{4}{3}$	32	Full projected area ratio for 4 blades = log (7) + 0.12483 = log (32).	log

	Form No. 2.	Form No. 3.	Form No. 4.
.....	log (16)	log (16).....	log (16).
.....	log (17)	log (17).....	log (17).
18)	.....	log (18)=log (15).....	log (18)=log (15).
.....	.....	log (19)	
log (20)	.....	log (20)=log (19).....	
.....	log (21)	log (21).....	log (21).
.....	.....	log (22)	
.....	.....	log (23)	
.....	log I.H.P. (estimated)=log (24).....	log (24) as in Form 1.....	log I.H.P. (esti.) = log (24).
.....	.....	log (23)—log (21)=log (25).	
21) — g (26) }	.....	log (26) as in Form 1.....	
.....	$\frac{1}{2}\{2.46509+\log(24)+2\log(21)-(\log(17)+\log(8))=\log(27)$.	.....	log (27) as in Form 2.
instead	.....	log (28) as in Form 1.....	
.....	For 4-bladed screws use log (29) instead of (27).	.....	log (29) as in Form 2.
; (1)=	0.49715 + log (17) + log (27) — log (1) =log (30).	log (30) as in Form 1....	log (30) as in Form 2.
(31)....	log (1)—(0.49715+log (27))=log (31)...	log (31) as in Form 1.....	log (31) as in Form 2.
blades	log (32)	log (32) as in Form 1.....	log (32) as in Form 2.


CORRECTIVE CHART FOR BLOCK COEFFICIENT WHERE B/L.W.L. OF ANY VESSEL DIFFERS FROM STANDARD B/L.W.L.
ALSO CORRECTIVE MULTIPLIERS FOR APPENDAGE RESISTANCE.
X=LINE OF BLOCK COEFFICIENTS ON WHICH SLIP CHARTS ARE CONSTRUCTED SUITABLE FOR WING SCREWS.
Y=LINE OF BLOCK COEFFICIENTS FOR INNER SCREWS OF 4 SHAFT ARRANGEMENTS, ALSO FOR CENTER SHAFT AND FOR SINGLE SCREWS THE DEAD WOOD BEING CUT AWAY IN BOTH LATTER CASES.
Z=LINE OF BLOCK COEFFICIENTS FOR ALL SCREWS WORKING DIRECTLY IN WAKE OF AND CLOSE TO THE STERN POST.
CORRECTIVE MULTIPLIERS IN % OF APPENDAGE RESISTANCES AS OBTAINED FROM CHART #2.
-SCALE OF GREATEST IMMERSED BEAM + LENGTH ON LOAD WATER LINE = B/L.W.L.-
GUIDE-LINES FOR SLIP CHARTS WHERE L.W.L. DIFFERS FROM
SCALE OF BL



OF BLOCK COEFFICIENTS.

Chart 1.

Lines of Appendage Resistance for Various Values of $\frac{\text{Beam}}{\text{L. L.W.L}}$ of Hulls of Standard Fineness of Forward and After Bodies as per Chart -I

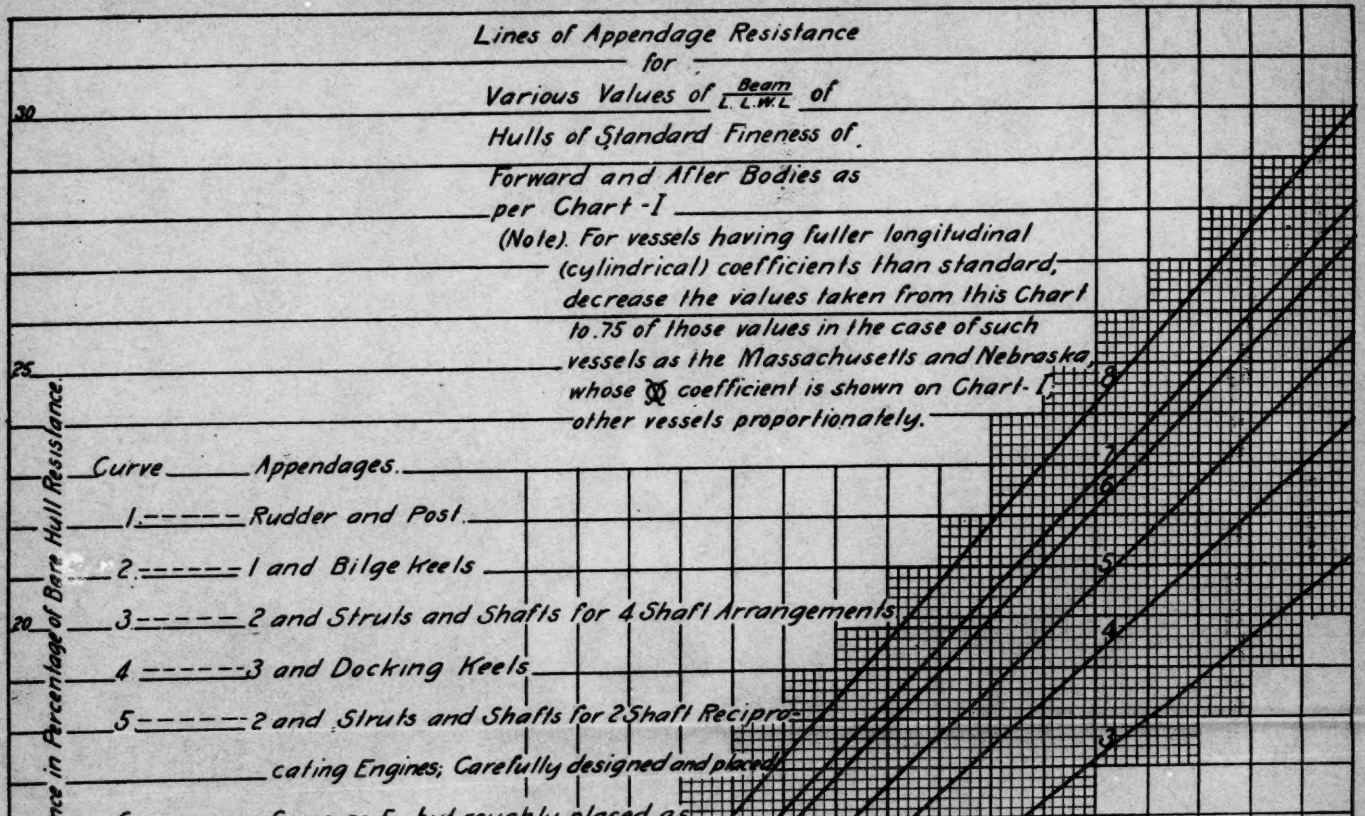
(Note). For vessels having fuller longitudinal (cylindrical) coefficients than standard, decrease the values taken from this Chart to .75 of those values in the case of such vessels as the Massachusetts and Nebraska, whose ⊠ coefficient is shown on Chart- I; other vessels proportionately.

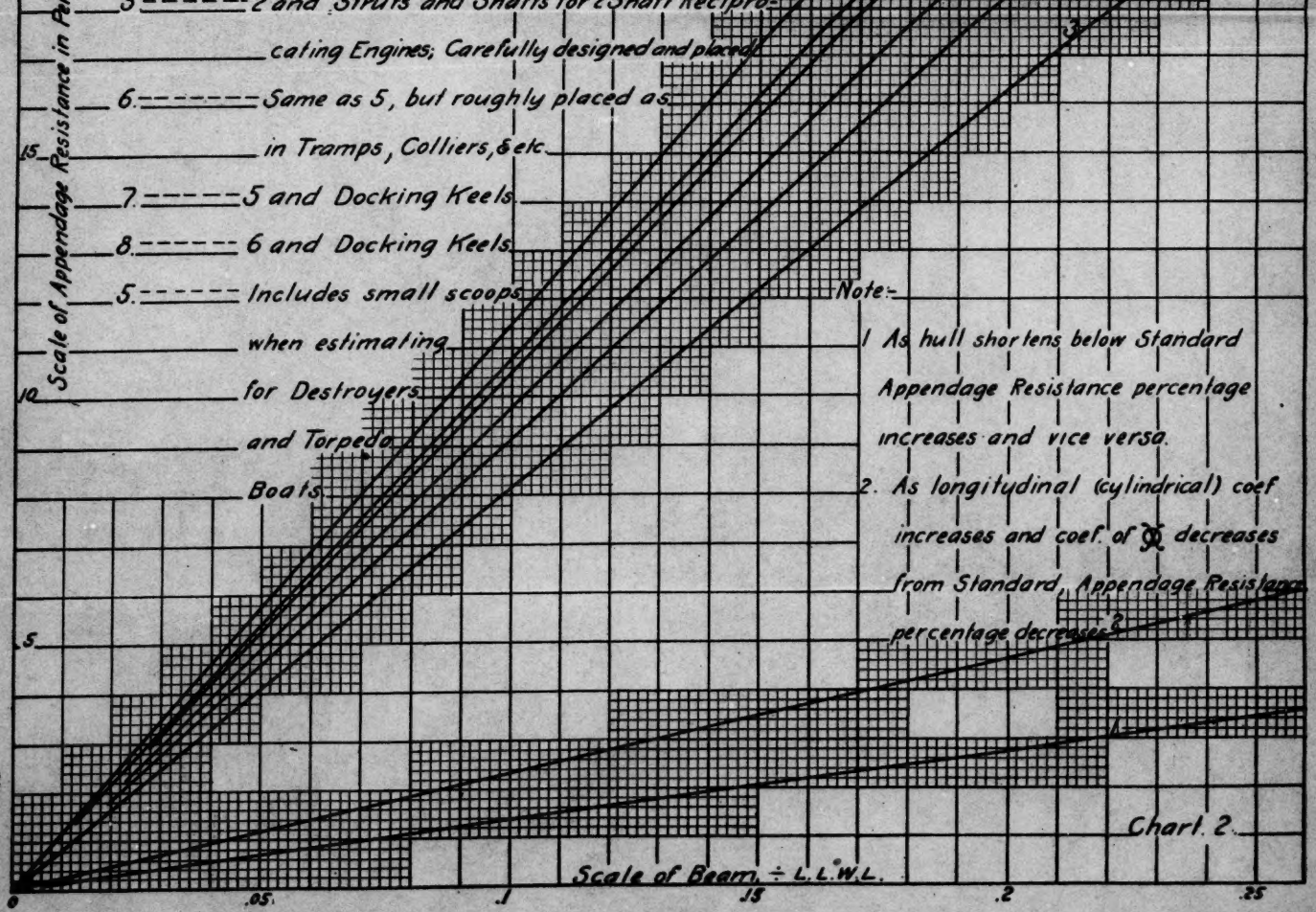
Curve ____ Appendages.

1. ------ Rudder and Post.
2. ------ 1 and Bilge Keels
3. ------ 2 and Struts and Shafts for 4 Shaft Arrangements.
4. ------ 3 and Docking Keels
5. ------ 2 and Struts and Shafts for 2 Shaft Reciprocating Engines; Carefully designed and placed.

Increase in Percentage of Bare Hull Resistance.

30
25
20




Scale of Appendage Resistance in Per
3 ------ 2 and Struts and Shafts for 2 Shaft Reciproc
cating Engines; Carefully designed and placed
6. ------ Same as 5, but roughly placed as
in Tramps, Colliers, & etc.
7. ------ 5 and Docking Keels.
8. ------ 6 and Docking Keels.
5. ------ Includes small scoops
when estimating
for Destroyers
and Torpedo
Boats
15
10
5
0
.05
.1
.15
.2
.25
Scale of Beam ÷ L.L.W.L.
Note:-
1 As hull shortens below Standard
Appendage Resistance percentage
increases and vice versa.
2. As longitudinal (cylindrical) coef
increases and coef. of ⊗ decreases
from Standard, Appendage Resistance
percentage decreases.
Chart 2.

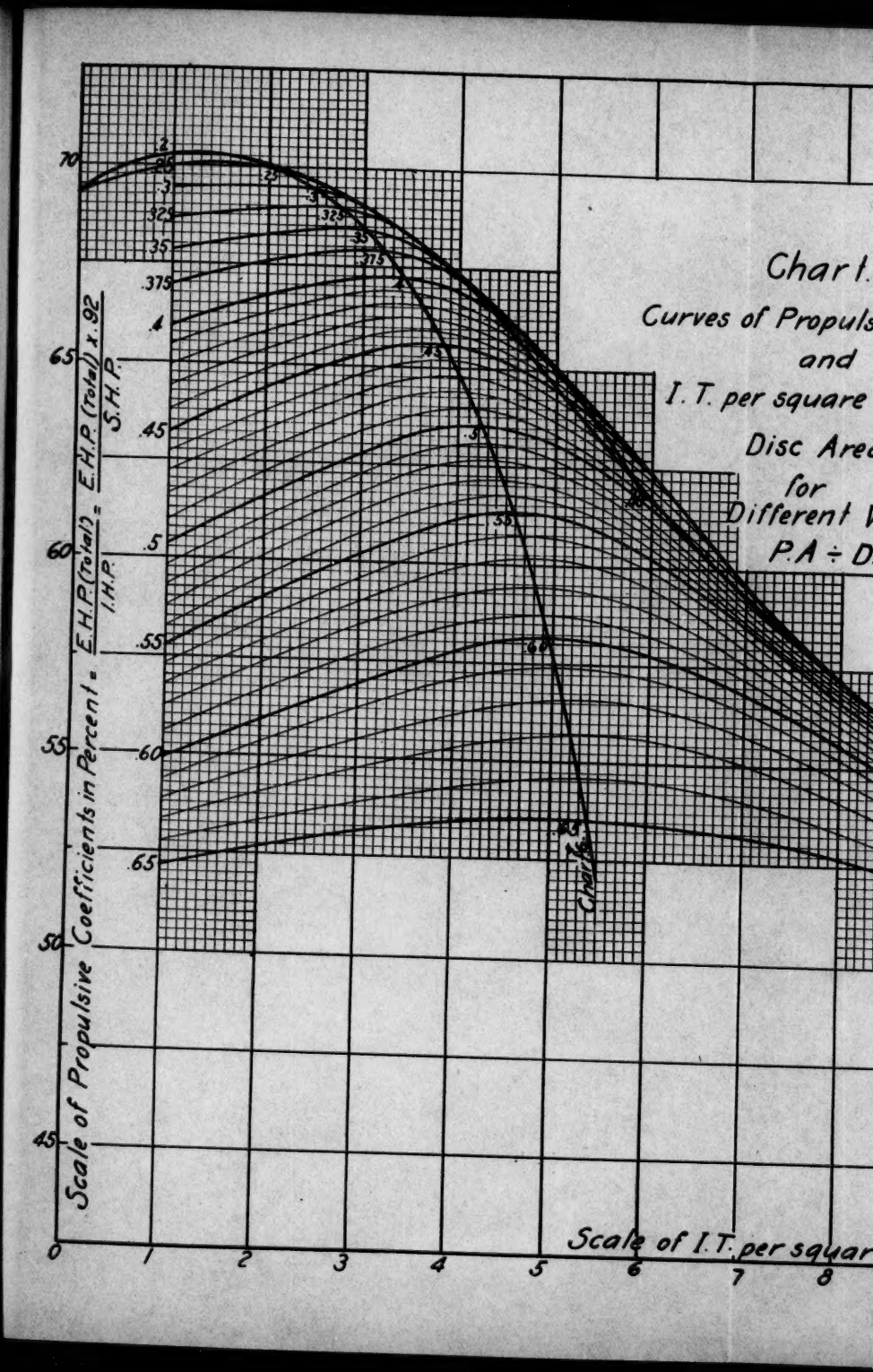


Chart I.
Curves of Propuls
and
I. T. per square
Disc Are
for
Different
P.A ÷ D
Scale of Propulsive Coefficients in Percent = E.H.P.(Total) / I.H.P. = E.H.P.(Total) x .92 / S.H.P.
Scale of I.T. per squar
Chart 2.
70
65
60
55
50
45
0
1
2
3
4
5
6
7
8
.2
.25
.3
.325
.35
.375
.4
.45
.5
.55
.60
.65

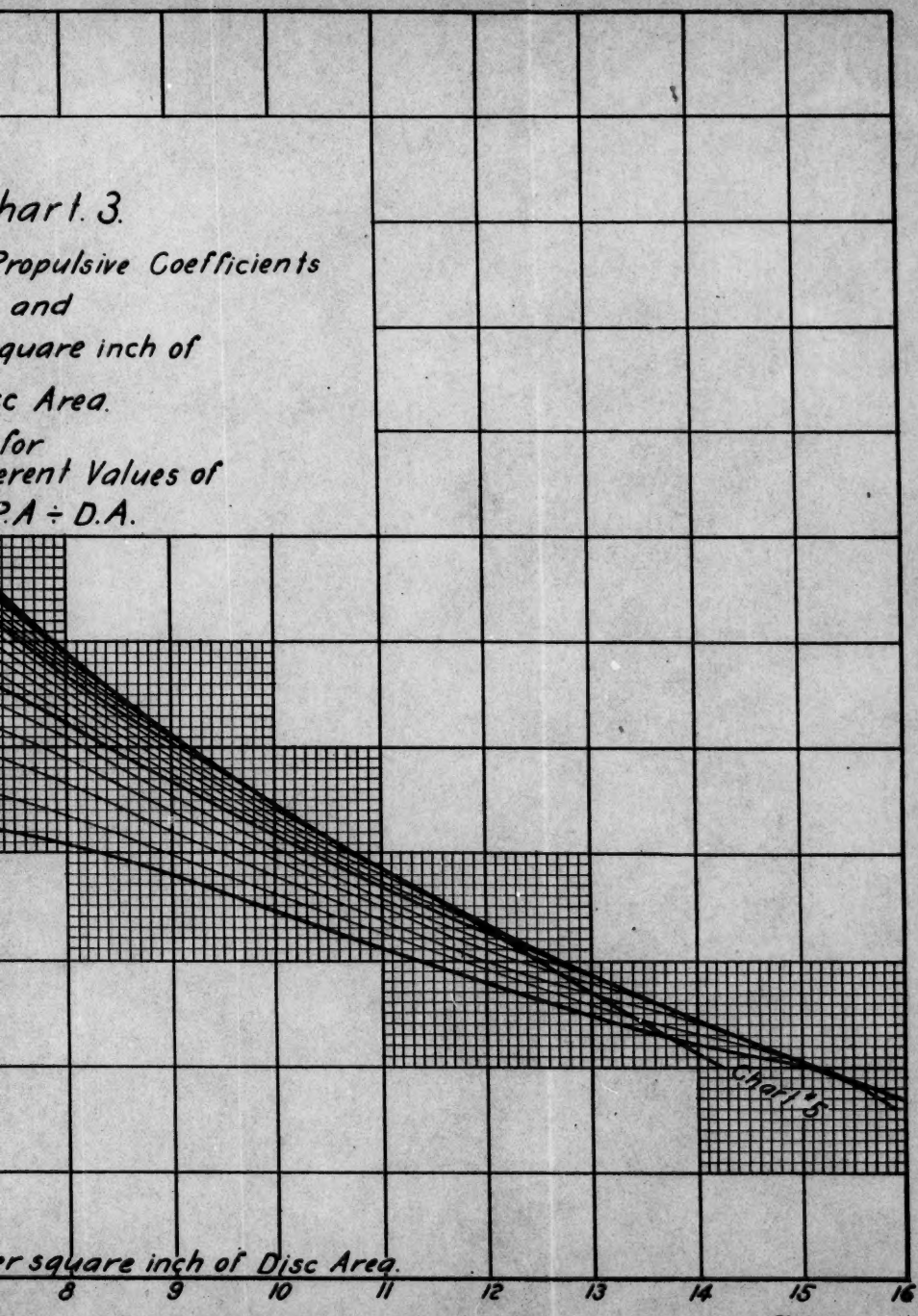

hart 3.
Propulsive Coefficients
and
quare inch of
c Area.
for
erent Values of
P.A ÷ D.A.
Chart 5
er square inch of Disc Area.
8
9
10
11
12
13
14
15
16

Chart 3.

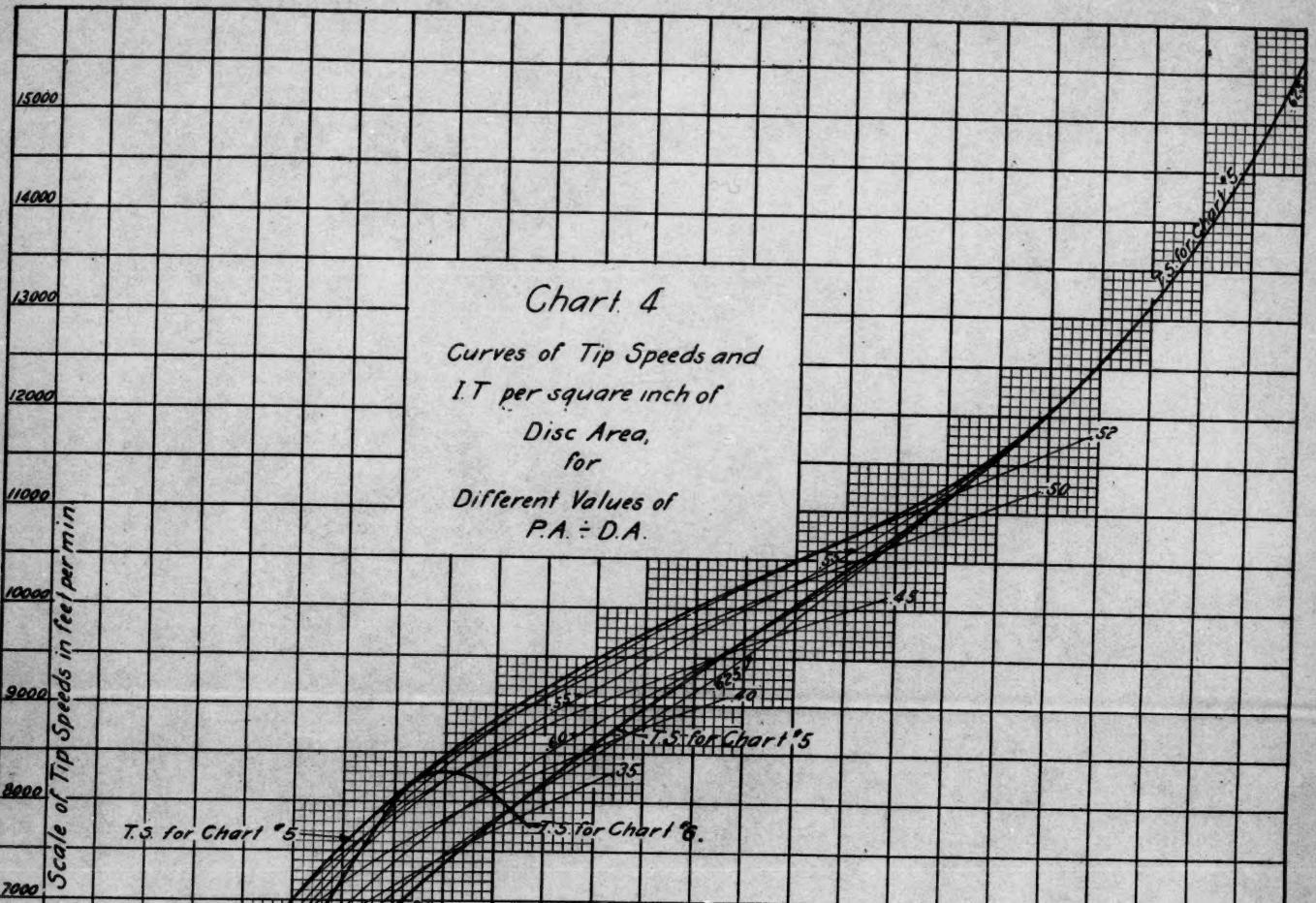
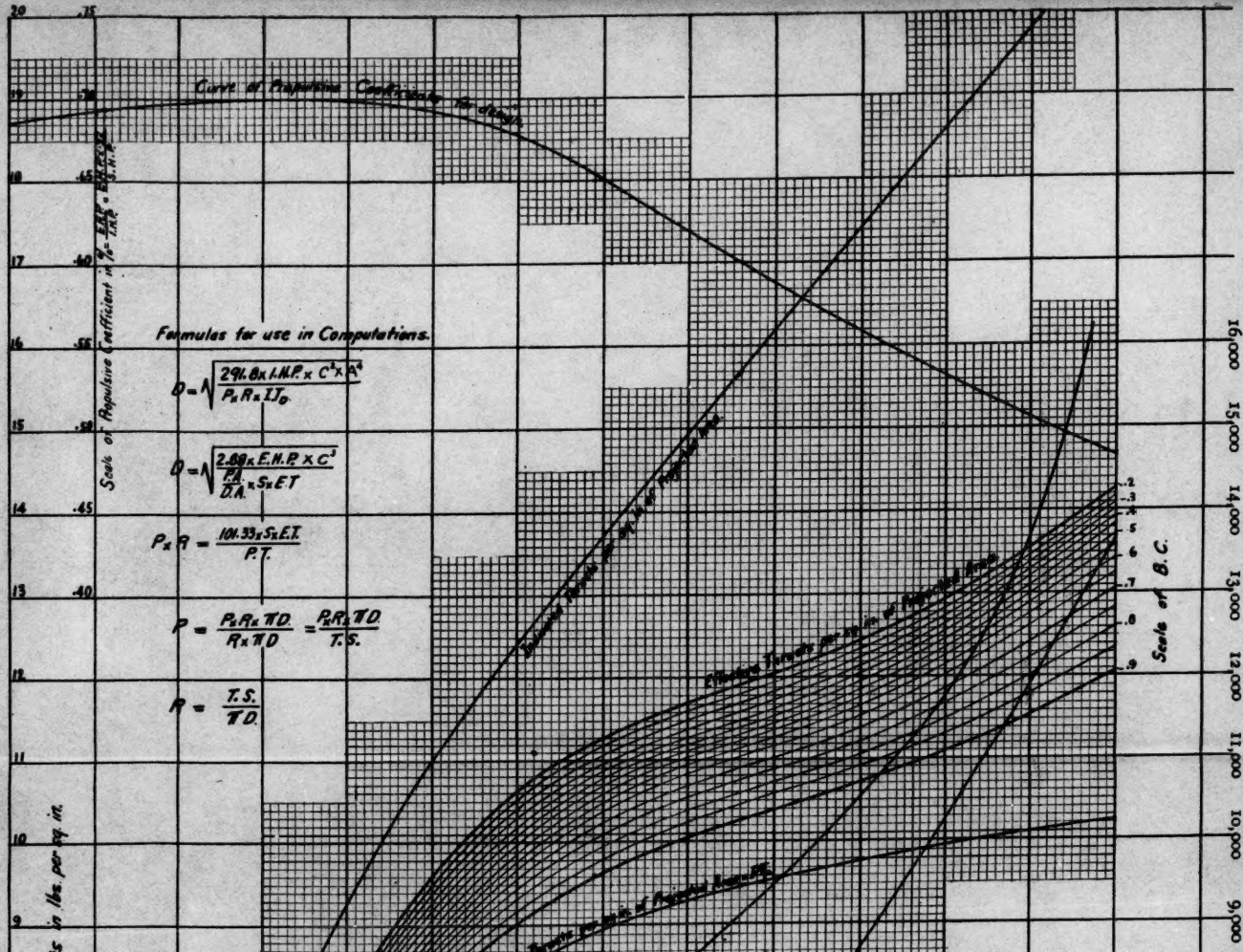


Chart 4
Curves of Tip Speeds and
I.T per square inch of
Disc Area,
for
Different Values of
P.A. ÷ D.A.
Scale of Tip Speeds in feet per min
15000
14000
13000
12000
11000
10000
9000
8000
7000
T.S. for Chart "5
T.S. for Chart "6.
T.S. for Chart "5
T.S. for Chart "5
.52
.50
.45
.40
.35

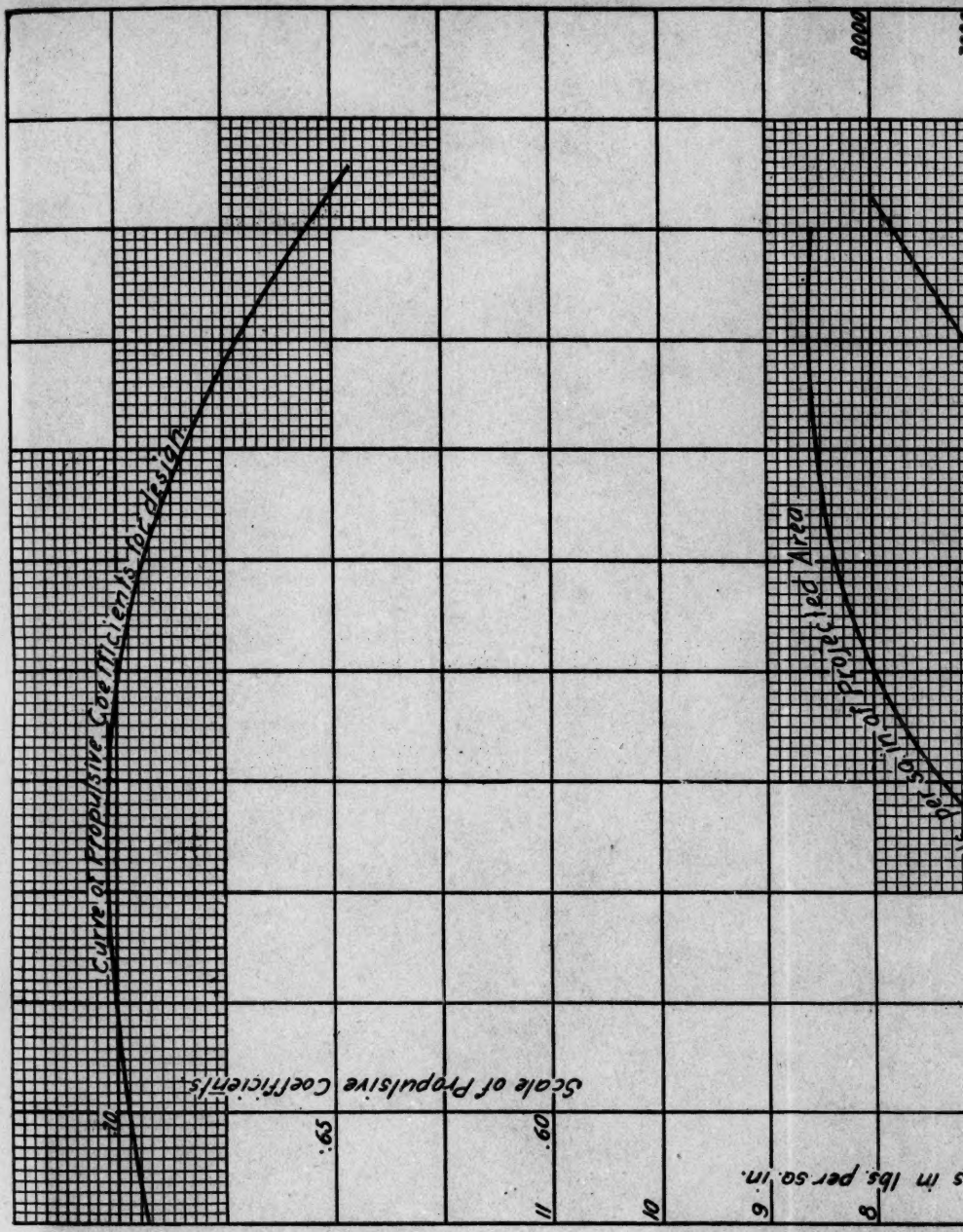
T.S. for Chart "5
T.S. for Chart "6.
Scale o
7000
6000
5000
4000
3000
2000
30
25
20
15
0
1
2
3
4
5
6
7
8
9
10
11
12
Scale of I.T. per square inch of Disc Area.

Chart 4.


Curve of Propulsive Coefficients for Steam
Scale of Propulsive Coefficient in % = E.H.P. / I.H.P. = E.H.P. / S.H.P.
Formulas for use in Computations.
$D = \sqrt{\frac{291.8 \times I.H.P. \times C^3 \times R^4}{P_s R_s I T_D}}$
$D = \sqrt{\frac{2.88 \times E.H.P. \times C^3}{\frac{P_A}{D_A} \times S \times E T}}$
$P_s R = \frac{101.33 \times S \times E.T.}{P.T.}$
$P = \frac{P_s R_s \pi D}{R \times \pi D} = \frac{P_s R_s}{T.S.}$
$R = \frac{T.S.}{\pi D}$
Absolute Pressure per sq. in. of Boiler Steam
Absolute Pressure per sq. in. of Propeller Blade Face
Scale of B.C.
Scale of Tip Speeds
Scale of Pressures in lbs. per sq. in.

Tip Speeds, in feet per minute.
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
Chart of Design for One and Two Shaft Arrangement.
Curves of
Tip Speeds and P.A./D.A.
I.T. per sq. in. of Disc Area and P.A./D.A.
I.T. " " " Proj. " P.A./D.A.
P.T. " " " " P.A./D.A.
E.T. " " " " P.A./D.A.
Propulsive Coefficients " P.A./D.A.
Ratio of P.A ÷ D.A.
Scale of Thrusts in lbs. per

Chart 5.


Curve of Propulsive Coefficients for Design
Scale of Propulsive Coefficients
Developed Area
Developed Area in sq. in.
Thrust in lbs. per sq. in.

CURVES OF PROPELLER

For vessels of low power and low speeds which are required to have maximum efficiency at constant

Chart 6.

OPELLER DESIGN.

t constant maintained sea speed. Suitable for tugs, colliers, freighters, tramp steamers, etc.

CURVES OF CORRECTION FACTORS, C, C^2 AND C^3,

FOR

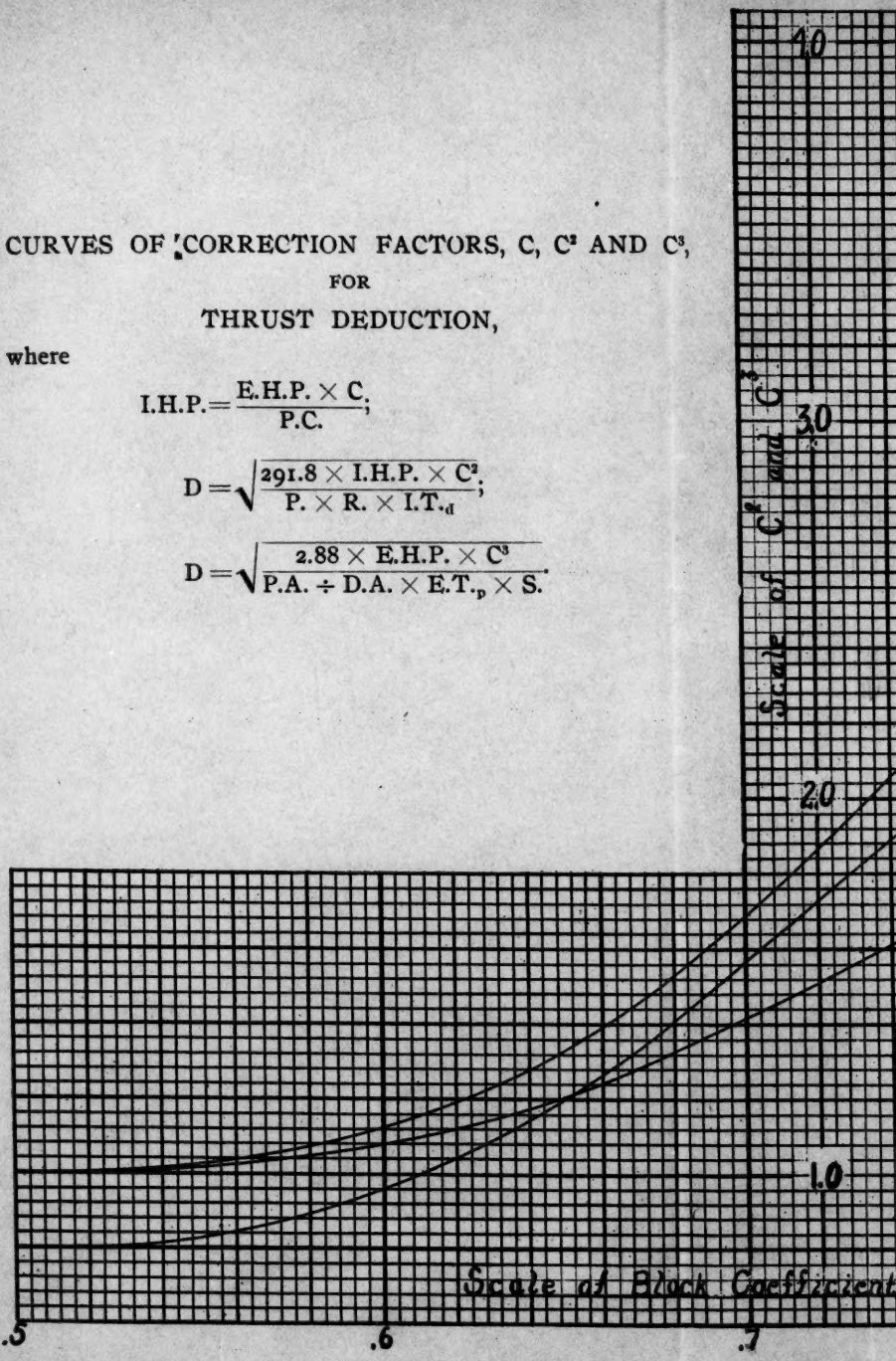
THRUST DEDUCTION,

where

$$\text{I.H.P.} = \frac{\text{E.H.P.} \times \text{C}}{\text{P.C.}};$$

$$\text{D} = \sqrt{\frac{291.8 \times \text{I.H.P.} \times \text{C}^2}{\text{P.} \times \text{R.} \times \text{I.T.}_d}};$$

$$\text{D} = \sqrt{\frac{2.88 \times \text{E.H.P.} \times \text{C}^3}{\text{P.A.} \div \text{D.A.} \times \text{E.T.}_p \times \text{S.}}}.$$



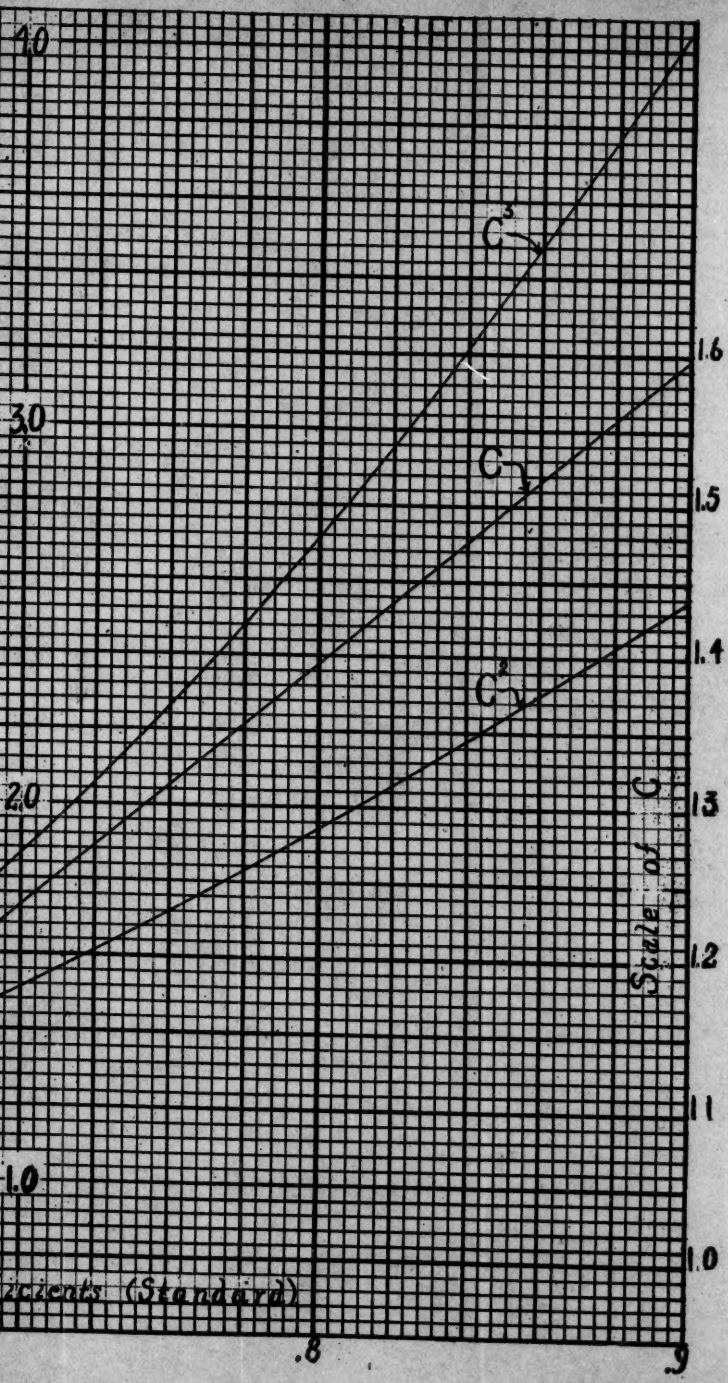


Chart 7.

DYSON STANDARD PROJECTED AREA

FOR

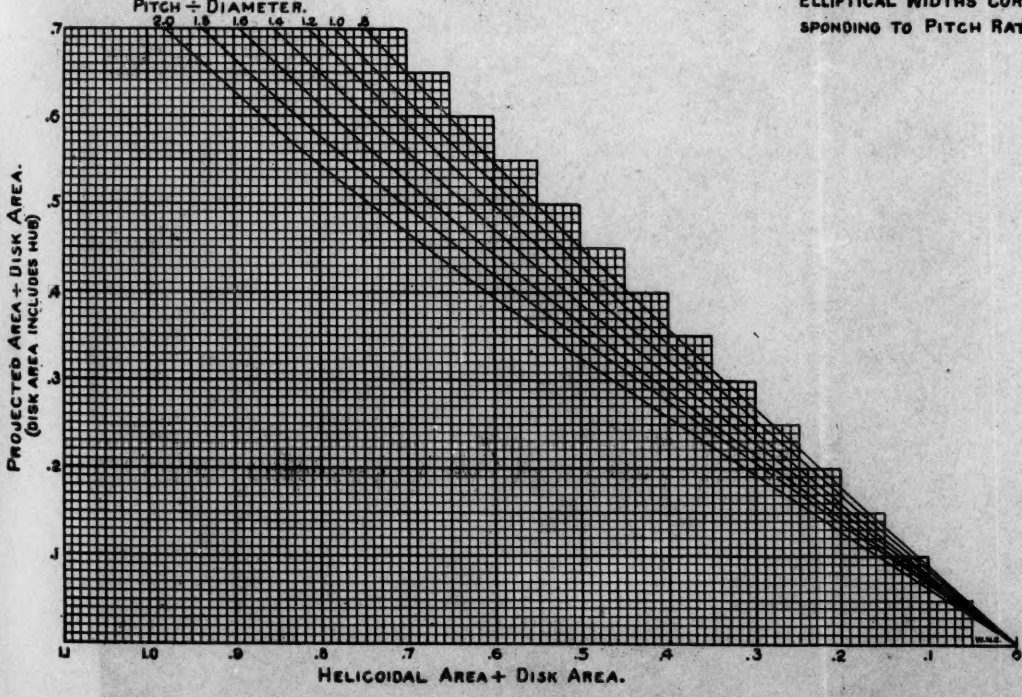
SCREW PROPELLERS WITH 3-4 AND 2 BL

JANUARY 1913.

TABLE OF HALF CHORDS, C., RADIUS-UNITY.

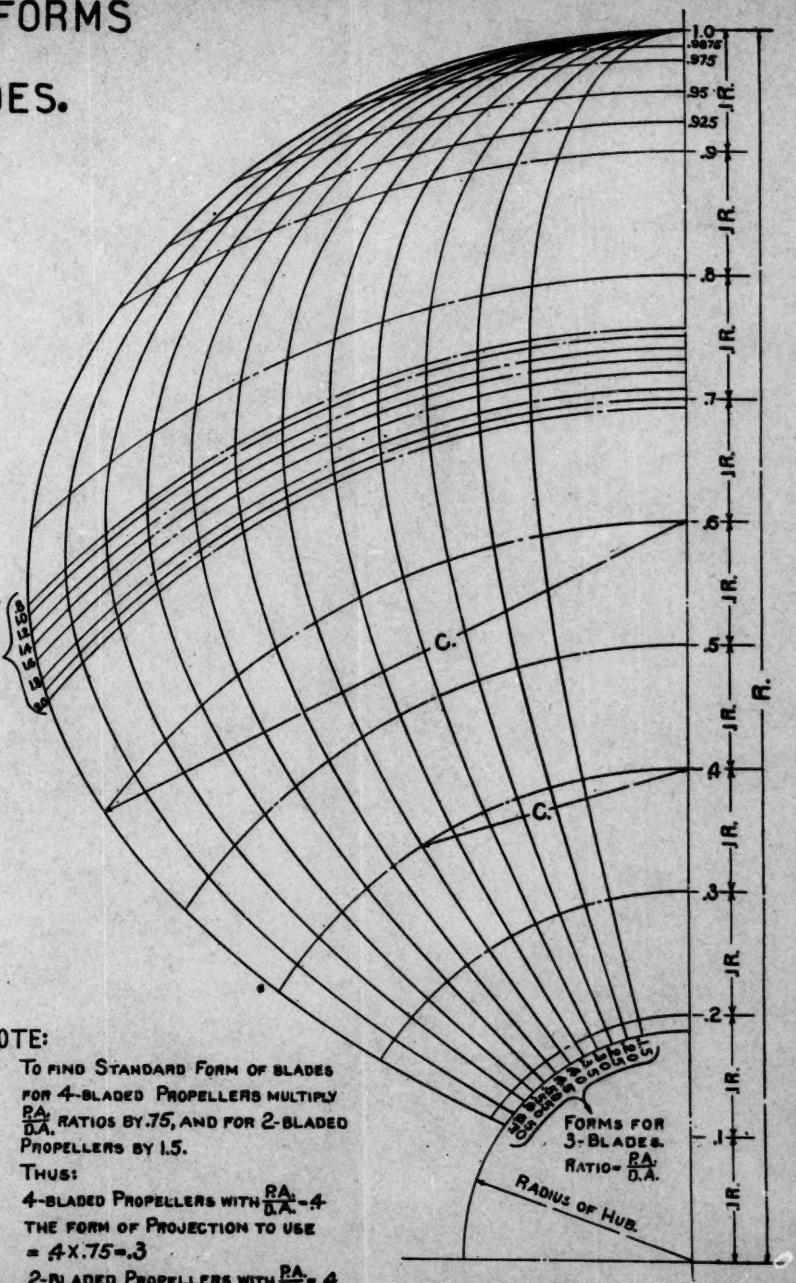
P.A.	DECIMAL PART OF RADIUS WITH HALF CHORDS UNDER.											
D.A.	.2	.3	.4	.5	.6	.7	.8	.9	.925	.95	.975	.9875
.15	.042	.064	.087	.107	.120	.128	.126	.105	.094	.080	.059	.042
.20	.055	.084	.114	.140	.159	.171	.168	.139	.125	.108	.078	.058
.25	.068	.107	.142	.173	.198	.212	.208	.174	.157	.133	.097	.069
.30	.082	.128	.170	.207	.236	.253	.250	.210	.190	.161	.120	.088
.35	.097	.150	.199	.242	.274	.295	.293	.245	.222	.192	.142	.105
.40	.109	.170	.227	.277	.316	.338	.333	.278	.251	.214	.158	.118
.45	.123	.191	.253	.308	.352	.378	.374	.313	.284	.243	.178	.127
.50	.136	.210	.279	.341	.389	.417	.415	.346	.312	.268	.195	.143
.55	.148	.230	.306	.372	.425	.459	.455	.382	.348	.293	.216	.165
.60	.159	.248	.332	.405	.463	.499	.497	.413	.377	.322	.238	.184
.65	.172	.270	.359	.437	.497	.539	.537	.445	.403	.343	.255	.200
.70	.184	.288	.382	.466	.533	.578	.575	.481	.437	.375	.280	.220

PROJECTIONS OF GREATE
ELLIPTICAL WIDTHS COR
SPONDING TO PITCH RA



AREA FORMS

2 BLADES.



OF GREATEST WIDTHS CORRE- PITCH RATIOS.

NOTE:

TO FIND STANDARD FORM OF BLADES FOR 4-BLADED PROPELLERS MULTIPLY $\frac{PA.}{DA.}$ RATIOS BY .75, AND FOR 2-BLADED PROPELLERS BY 1.5.

THUS:

4-BLADED PROPELLERS WITH $\frac{PA.}{DA.}$ = .4 THE FORM OF PROJECTION TO USE = .4 X .75 = .3

2-BLADED PROPELLERS WITH $\frac{PA.}{DA.}$ = .4 THE FORM OF PROJECTION TO USE = .4 X 1.5 = .6.

Chart 8.

PHOTOMICROGRAPHIC EXAMINATION OF BROKEN CRANK SHAFT.

To determine, if possible, the cause of failures in submarine crank shafts, a photomicrographic examination was made of two broken crank shafts of the C–4. This examination was made by Lieutenant J. F. Daniels, U. S. Navy, at the Naval Engineering Experiment Station, Annapolis, Md.

The specimens first submitted were taken from the (b) shaft, near the point of rupture, and were not of sufficient size to furnish specimens for physical tests, but were ample for microscopic purposes.

The following are the results of the physical tests when the shaft was manufactured:

(a) Chemical Analysis: $P = .007$; $S = .022$; $C = .41$; $Mn = .32$; $N = 3.27$.

(b) Heat treatment: Oil tempered and annealed twice.

(c) Material: Nickel steel, Class A Machinery Forging.

Diam. of test piece.	Elastic limit.	Tensile strength.	Elong. pr. ct.	Reduction pr. ct.	Area of fracture.
.505	47,500	75,250	31.5	65.86	.0683
.505	54,500	85,500	25.9	60.58	.0789
.505	54,500	88,500	24.5	54.13	.0919
.505	53,000	87,000	27.0	59.08	.0819
.505	50,000	83,000	24.65	48.87	.1024
.505	49,000	82,500	23.7	46.60	.1069

Photomicrographs.—The photomicrographic figures selected are shown in figures numbered 1 to 15, inclusive. Figs. Nos. 1 to 11 are described below.

Fig. No. 1. Represents the microstructure of a specimen

taken near the point of rupture. A continuous crack penetrated the metal in this section, the crack following, for a portion of its length, the cell walls of ferrite which enclose the pearlite walled cells. Near the bottom of the figure the crack passes directly through the lower walled cell shown.

The walled cells are of enormous size, giving evidence of high temperatures during heat treatment. Small slag enclosures are shown imbedded in the ferrite walls in the extreme left-hand side of the figure.

Fig. No. 2. Contains a figure showing the microstructure of a specimen taken from a region close to the rupture. The figure was selected to show the typical crystallographic formation found generally, and also the form of manganese sulphide inclusions characteristic of the metal under investigation.

Fig. No. 3. Shows the characteristic network common to the metal under investigation. Extending vertically throughout the figure is a continuous slag line. Smaller particles of slag are seen both to the right and left of the central streak.

Fig. No. 4. Contains a photomicrograph taken from a polished specimen. This figure shows a continuous heavy slag line, and several outlying smaller patches.

Fig. No. 5. Represents the crystallographic formation of a region close to that pictured in Fig. No. 4. Continuous plates of slag are clearly shown.

Figs. Nos. 6 and 7. Show polished surfaces. The figures were taken close to the rupture and give evidence of continuous slag enclosures.

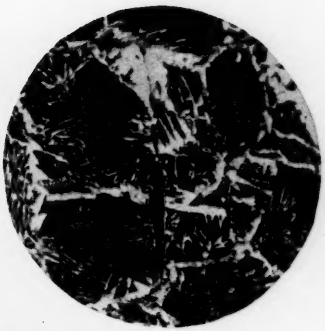
Figs. Nos. 8 and 9. Contain etched figures. Continuous plates of manganese sulphide are shown.

Fig. No. 10. A polished figure showing plates of impurities.

Fig. No. 11. This photomicrograph was selected to illustrate the fact that certain forms of sulphide and silicate inclusions are extremely harmful in that they form planes of weakness from which cracks may emanate. The figure shows such a condition, a fine crack having developed at the lower end of one of the slag lines shown and extends downward from it.

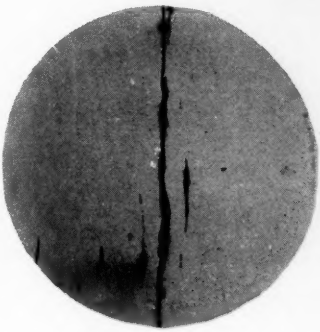
The photomicrographic figures described above were taken

1



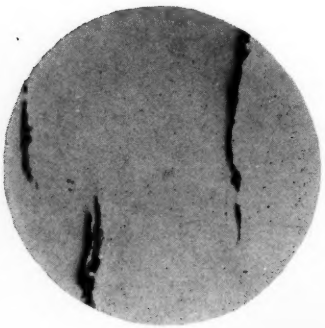
2

3

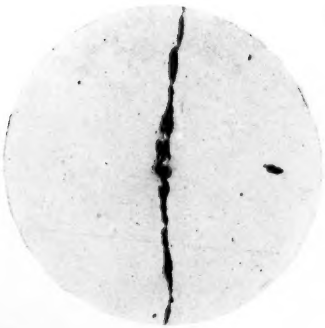


4

5



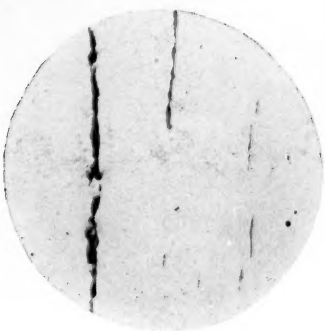
6



7

8

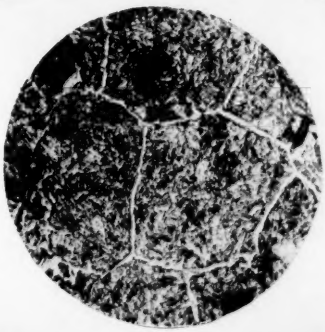
9



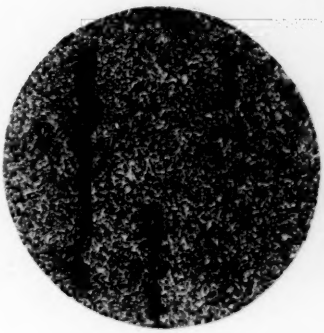
10

11

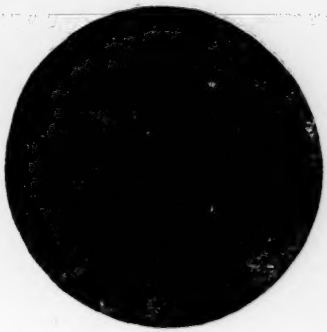
12



13



14



15

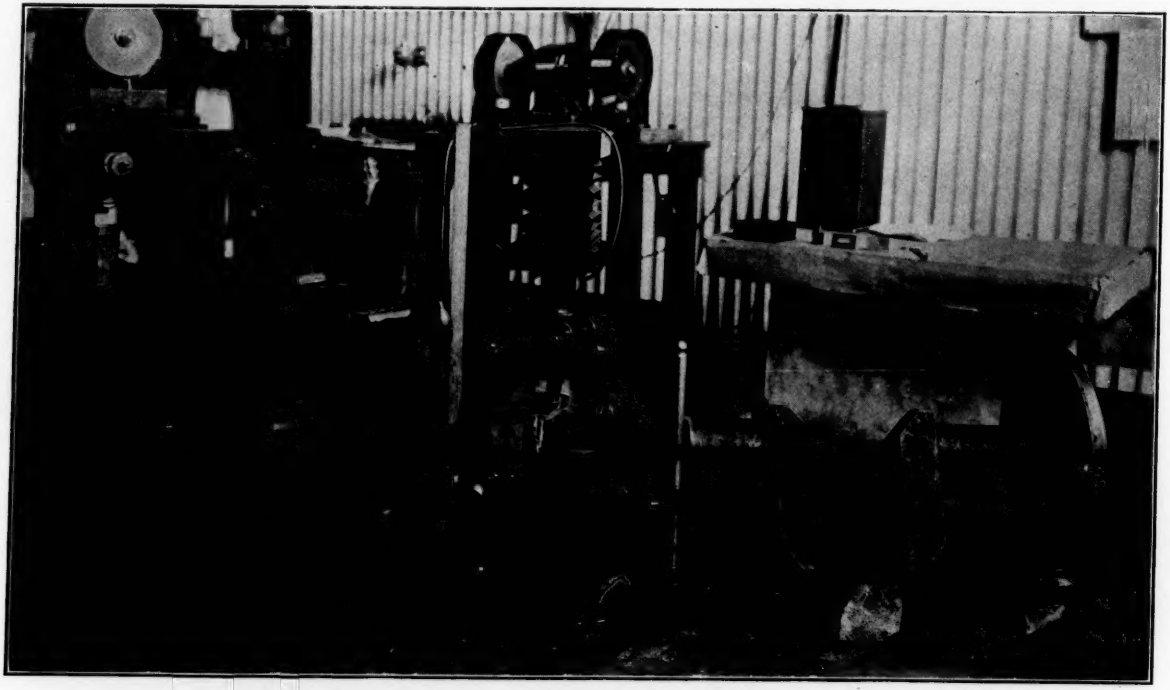


PLATE I.—SHOWING APPARATUS MOUNTED ON UNIVERSAL STAND.

from small specimens cut from the sample submitted for investigation. During the progress of the work it was found that the metal under examination presented unusually good material for use in connection with repeated stress investigations, and a request that the whole crankshaft be sent to the Engineering Experiment Station was submitted. In due time this shaft was received, but, on examination, the shaft proved to be the one designated as (a), instead of the one designated as (b).

That it was not the shaft already under investigation did not become apparent immediately. In continuing the examination with the supposedly unlimited material now at hand, it was decided to prepare and photomicrograph a section of the shaft proper, rather than photomicrograph specimens cut therefrom. A section of the web was polished by hand and then etched with a 5 per cent. solution of nitric acid in alcohol. The microscope was mounted on a universal stand as shown in photograph, Plate I, and a microscopic examination made of the etched region. Three important facts were determined: (1) that it was not the same shaft under investigation, (2) that the heat treatment was unsatisfactory, and (3) that it contained harmful forms of sulphides and silicates.

The results of this examination are shown in figures Nos. 12 and 13:

Fig. No. 12. This photomicrograph shows a great streak of sulphide extending across the figure from top to bottom. The wall cells, as in the previous examination of the (b) shaft, were abnormally large. The cell walls of ferrite were much narrower.

Fig. No. 13. This figure was taken from a region close to the above and shows a slag area in the form of patches rather than of streaks.

These figures are characteristic of the crystallographic structure of the metal in the (a) shaft generally.

With a view to determining the effect of reheat treatment on the metal of the (b) shaft, a number of specimens were heat treated in a small oil furnace.

Beginning with a temperature of 550 degrees C., and carrying it up successively to 1,100 degrees C., a number of points were obtained as follows:

Piece.	Temp.	Cooled.	Structure.
1	550° C.	In furnace to 350° C., then in air.	No change in structure.
2	650° C.	In furnace to 500° C., then in air.	No change in structure.
3	750° C.	In furnace to 500° C., then in air.	Structure changed from cellular, or network, to a homogeneous, grainlike structure.
4	950° C.	In furnace to 500° C., then in air.	Structure changed slightly, but still of cellular, or network, variety. The network size slightly smaller than originally.
5	1,100° C.	In furnace to 500° C., then in air.	Coarse network identical with original structure.

Photomicrographic records were made of pieces Nos. 3 and 4. The crystallographic structures developed are shown in figures Nos. 14 and 15, No. 14 having the granular, and No. 15 the network, structure. The heat-refining process has broken up the coarse network, but the impurities are still in evidence since heat treatment cannot remove these.

The shaft is said to have been "oil tempered and annealed, twice." No mention is made of the heat factors or cooling periods used in the method of treating. *These are essential factors* to produce proper results. The treatment received in the case of both the (a) and (b) shafts shows the heat factors to have been too high to produce a close-grained homogeneous metal.

To properly heat-treat steel it is necessary to determine the thermal critical points of the metal if the relation existing between treatment and structure is to be observed.

Steel treated to simply pass the physical tests required may be found satisfactory for ordinary members in mechanical

construction, but for important members, such as crankshafts, etc., the relation existing between structure and physical properties should receive the most careful attention.

Quoting from "Mellor":

"The physical properties of steel may be taken to depend upon—

"(1) Chemical composition, *i. e.*, the relative amounts of other elements present;

"(2) Distribution of constituents, *i. e.*, the relative proportions of ferrite, cementite, etc., present;

"(3) Size of grains."

A few tests conducted by "Sauveur" on the relation between the size of the grain and the physical properties of the same piece of steel, are shown below:

Size of grain, in 0.0001 sq. mm.	Tensile strength, kilogrm. per sq. mm.	Elongation, pr. ct. of length.	Reduction, pr. ct.
148	69.6	15.0	20
118	70.3	19.0	22
62	77.7	22.5	35

From the above it would seem that the smaller the grain size the better will be the steel.

In order to determine the proper heat factor to apply in connection with the heat treatment of a particular metal, the thermal critical points should be located and the treatment proceed in accordance with information thus established. Indeed, having heat-treated an important member, a microexamination should be made of the finished product, selecting several places on the shaft, in order to ascertain whether or not the whole piece has received uniform treatment. Finding the piece unsatisfactory as regards structure will not necessarily condemn it, since heat refining is capable of bringing about satisfactory results.

For an excess of impurities there is no cure. This must be considered under head of prevention during manufacture, and

should receive earnest attention. After the furnace is tapped a sufficient time should elapse for impurities to arise in order that they may be skimmed off, or permitted to overflow into cinder pit. When pouring into ingot molds clear metal will flow through bottom of ladle provided the nozzle therein is not too large. With large nozzles, especially when the molten level within the ladle is now, the swirling or whirlpool action of the mass may pull impurities, not taken off by skimming or overflowing prior to pouring, into the mold.

Tests.—The strength of a metal when tested for tensile and bending properties, is not a complete guide if the metal is to be subjected to a series of constantly vibrating stresses. Metals are said to " fatigue," and when in this condition will break more readily.

Repeated stress testing takes up more time than the manufacturer is willing to utilize, as a rule, and in consequence such testing must be conducted in the laboratory.

Three contributory causes are suggested by this examination: (1) excess of impurities, (2) poor heat treatment, and (3) fatigue.

The impurities created weak planes in the material, making it possible for rupture to follow across the intervening bridges of normal metal from weak plane to weak plane.

The abnormally large net work unbalanced the metal. Quoting from " Howe" (Structure of Steel, Paper, " Proceedings of American Society for Testing Materials," Vol. XI, 1911):

" Let us assume there is a well-marked network structure. In the case of hypo-eutectoid steel these (cell) walls have more iron than the kernels (walled cells), and they lack carbon nearly or quite completely. The process of equalizing (heat refining, or grain refining) then necessarily implies—1, the diffusion of enough carbon from within the kernels into the cell walls to give the whole a uniform carbon content; and, 2, a simultaneous outflow of enough iron from those cell walls into the kernels to give a uniform iron content throughout."

As to fatigue, it would appear from recent practice that chrome-vanadium steels have a higher value than nickel-steels when compared from an anti-fatigue viewpoint.

It is recommended that in the case of all important members, such as crankshafts, piston rods, etc., a microphotographic record be made of the finished product, and these be filed as a part of the card history of the material.

Quoting from "Sauveur":

"It is to the close relation existing between the treatment and the structure on the one hand, and between the structure and the physical properties of the metal on the other, that metallography owes its industrial importance."

The importance of metallography, since it reveals the structure, should be recognized; and the card history of important steel members should be made complete in respect to this science.

Inspectors should be qualified to make micro-inspections of important steel members. This would not necessarily require a photographic record, but would insure the proper heat treatment in the case of the material under inspection.

There is no difficulty attached to making such an inspection, and none to making the photographic record if such be required.

TEST OF A GARY-CUMMINGS TORSION METER FOR THE U. S. S. *DUNCAN* AT THE WORKS OF THE FORE RIVER SHIPBUILDING COMPANY.

By Lieutenant (J. G.) Warren G. Child, U. S. Navy.

Briefly the Gary-Cummings torsion meter consists of a tube, the length varying for different shafts, installed within a section of line shafting, one end secured within the shaft by pointed set screws, the other end, carrying the pointer mechanism, riding free within a ball bearing at the flange of the shaft. A lever which actuates the pointer mechanism passes through a slot in the flange, the slot being large enough to permit the free movement of the lever within the radius of movement of the pointers. With no load on the shaft the two pointers, when correctly adjusted, lie in the same plane of rotation, but as the load varies the pointers separate a distance directly proportional to the twist of the shaft. A recording apparatus makes it possible to obtain at any time this distance.*

The object of calibrating the torsion meter is to determine the relation existing between the movement of the pointers and the horsepower transmitted by the shaft.

The calibration test described was for the port shaft of the U. S. S. *Duncan.* The method pursued was as follows (see Plates I, II and III):

The forward end of the shaft was rigidly secured to a heavy casting clamped to the bed of a large boring machine. The after end rode freely in a ball bearing just abaft the flange, likewise rigidly secured. To the after flange was bolted a lever, X, Y, with ten-foot arms, for applying the torque. Square marks were placed on the flanges to show any slipping of the shaft or lever. The two pointers, A, B and C, D

* See Journal A. S. N. E., May, 1912, for full description of torsion meter.

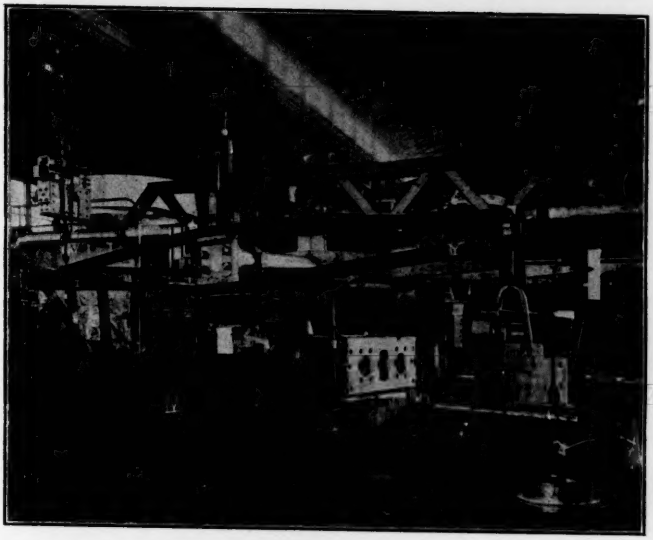


PLATE I.

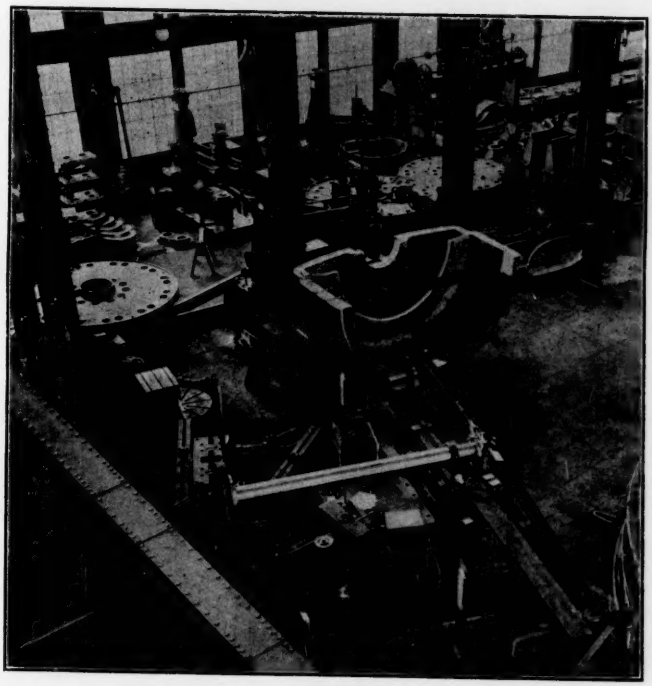


PLATE II.

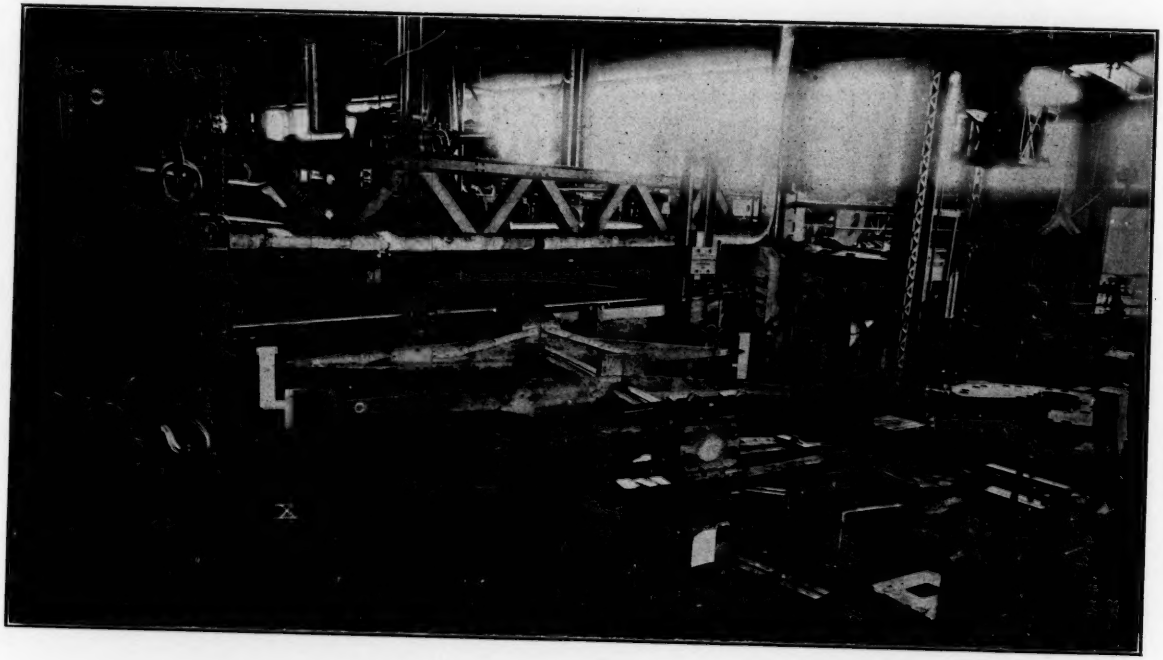


PLATE III.

(eight-foot arms), for determining the angle of twist were clamped securely to the shaft six feet apart. All measurements were accurately checked. Cards were secured in position at A, B, C and D for recording the pointer readings, and on small boards for taking the meter readings. Squares and sharp pencils were provided for marking pointer positions. Beam scales were placed in position at each end of the lever, suspended from a crane at one end and secured to heavy weights at the other. The upward pull was obtained by means of a differential purchase, the downward pull by a bolt and nut. The pull on the lever was taken on knife-edge bearings at each end.

The moment required to produce an overload of 25 per cent. above the designed horsepower of the shaft was determined from the formula, $M = \frac{S.H.P. \times 33{,}000}{2\pi R.p.m.}$. This result was used in calculating the pull to be applied to the lever arms.

At a bell signal the zero positions were recorded with no load on the lever.

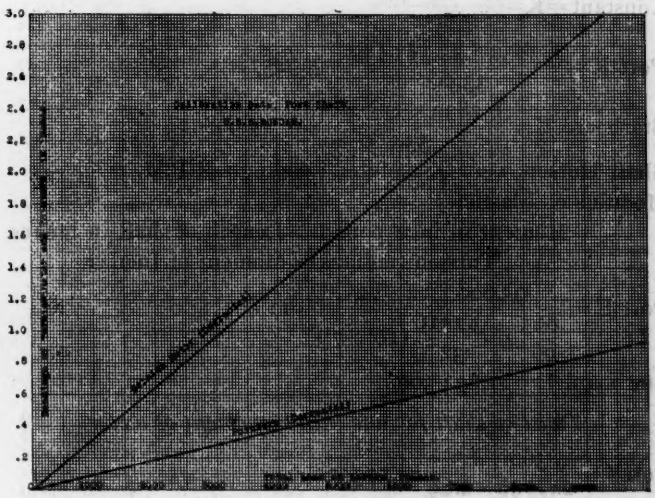


FIG. 1.

Then a load of approximately 500 pounds was placed on each arm, the bell was rung, the figure "1" was displayed to denote the number of the reading, and pointer positions were again marked.

Seven readings were taken in this manner with increasing loads, then seven with decreasing loads. The loads on the lever arms were kept as even as possible, the discrepancy in the readings given in columns X and Y being due to the actual weight of the scales, which acted as a downward pull.

The formulas and observed data are as follows, the corrected results being shown in Fig. 1.

$$\text{Slope} = \frac{\text{Moment}}{\text{Meter Reading}} = \frac{\text{Weight} \times \text{Arm}}{\text{Meter Reading}}.$$

Port slope

$$= \frac{9150 \times 10}{2.94} = 31{,}122.449 \text{ ft. lbs. per inch of meter reading.}$$

Starboard slope

$$= \frac{9000 \times 10}{2.9} = 31{,}034.483 \text{ ft. lbs. per inch of meter reading.}$$

$$\text{Constant} = K = \frac{2\pi \times \text{Slope}}{33{,}000}.$$

$$\text{Port } K = \frac{6.2832 \times 31{,}122.449}{33{,}000} = 5.9257.$$

$$\text{Starboard } K = \frac{6.2832 \times 31{,}034.483}{33{,}000} = 5.9089.$$

Shaft Horsepower $= K \times$ Meter Reading $\times$ R.P.M.

Modulus of Elasticity

$$= E = \frac{\text{Moment in inch pounds} \times \text{Length of shaft in inches.}}{\text{Torsion (radians)} \times \text{Moment of Inertia}}$$

$$\text{Port } E = \frac{95{,}000 \times 12 \times 72}{\frac{.895}{96} \times \frac{3.1416}{32}(10^4 - 7^4)} = 11{,}801{,}3.00.$$

Starboard $E = 11{,}834{,}300.$

Twist of shaft for one inch movement of meter.

$$\text{Port } \theta = \frac{360 \times .85}{2\pi \times 96 \times 2.89} = .17553°.$$

Starboard $\theta = .17411°.$

CALIBRATION DATA, PORT SHAFT, U. S. S. "DUNCAN," GARY-CUMMINGS TORSION METER, METER NO. 22.

X = upward pull on ten-foot lever arm.
Y = downward pull on ten-foot lever arm.
Correction to be applied to weights = + 532.
A and B = after pointers, radius 8 feet.
C and D = forward pointers, radius 8 feet.
Distance between pointers = 6 feet.
* = upper readings corrected to 0.

Number.	Up, X.	Down, Y.	X + Y.	Correct, total.	Meter.	Corrected to.	A.	B.	C.	D.	Mean, A & B.	Mean, C & D.	Difference.
FIRST RUN.													
0	0	0	0	...	+.04	0.	0.	0.	0.	0.	...	...	...
1	500	135	635	1167	.33	.37	.03	.03	.15	.15	.03	.15	.12
2	1195	740	1935	2467	.74	.78	.10	.085	.32	.32	.09	.32	.23
3	1915	1360	3275	3807	1.18	1.22	.18	.16	.55	.53	.17	.54	.37
4	2640	2030	4670	5202	1.63	1.67	.27	.255	.75	.74	.26	.745	.485
5	3265	2730	5995	6527	2.08	2.12	.37	.35	.99	.97	.36	.88	.62
6	4005	3415	7420	7952	2.52	2.56	.48	.47	1.23	1.20	.475	1.215	.74
7	4470	3900	8370	8902	2.84	2.88	.59	.56	1.44	1.42	.575	1.43	.855
8	3960	3500	7460	7992	2.53	2.57	*.38 .56	...	1.15 1.33	1.13 1.30	.38	1.14	.76
9	3280	2775	6055	6587	2.07	2.11	*.33 .51	.34 .52	.97 1.15	.94 1.11	.335	.955	.62
10	2590	2090	4680	5212	1.64	1.68	*.28 .46	.27 .45	.77 .95	.76 .93	.275	.765	.49
11	1900	1370	3270	3802	1.20	1.24	*.22 .40	.22 .40	.58 .76	.57 .74	.22	.575	.355
12	1200	690	1890	2422	.75	.79	*.15 .33	.155 .335	.39 .57	.37 .54	.15	.38	.23
13	490	155	645	1177	.34	.38	*.09 .27	.09 .27	.19 .37	.19 .36	.09	.19	.10
14	0	0	0	...	+.03	.00	*.00 —.18	.00 —.18	.00 —.18	.00 —.17	...	...	...
SECOND RUN.													
0	...	...	0	...	+.03	0.	.00	0.	.00	0.	...	...	...
1	500	160	660	1192	.35	.38	.04	.04	.17	.175	.04	.17	.13
2	1165	725	1890	2422	.75	.78	.11	.11	.34	.345	.11	.34	.23
3	1915	1380	3295	3827	1.19	1.22	.18	.18	.55	.55	.18	.55	.37
4	2600	2105	4705	5237	1.64	1.67	.24	.24	.74	.74	.24	.74	.50
5	3345	2775	6120	6652	2.09	2.12	.32	.32	.94	.94	.32	.94	.62
6	4045	3460	7505	8037	2.53	2.56	.37	.37	1.14	.113	.37	1.135	.765
7	4100	4025	8125	8657	2.74	2.77	.41	.41	1.225	1.22	.41	1.22	.81
7A	4505	3985	8490	9022	2.86	2.88	.43	.405	1.27	1.27	.42	1.27	.85
8	3995	3500	7495	8027	2.53	2.55	.42	.38	1.14	1.13	.40	1.135	.735
9	3300	2790	6090	6622	2.09	2.11	.38	.34	.965	.95	.36	.955	.595
10	2595	2095	4690	5222	1.65	1.67	.33	.29	.78	.77	.31	.775	.465
11	1885	1390	3275	3807	1.19	1.21	.22	.22	.58	.57	.22	.575	.355
12	1205	700	1905	2437	.75	.77	.16	.16	.38	.37	.16	.375	.215
13	500	155	655	1187	.34	.36	.08	.08	.19	.19	.08	.19	.11
14	0	0	0	...	+.02	.00	.00	.00	.00	.00	...	...	...

SALT IN BOILER FEED WATER.

By Ensign E. E. Wilson, U. S. Navy.

In a large number of engineering plants both ashore and afloat the possibility of obtaining great economy through maintaining a low degree of salinity in the steaming boilers is not appreciated. It is understood everywhere that salt feed water, by depositing scale on the heating surface reduces appreciably a boiler's efficiency, but it is not so generally noted that a boiler steaming with absolutely fresh water has a vastly higher evaporative efficiency than when steaming with water containing even a relatively low quantity of salt. A study of this fact, together with a consideration of the source of salt in boiler feed as well as the methods of eliminating it, may assist in the attainment of greater economy in coal consumption.

It is well known that, with the small-tube boilers now in use, a large salt leak in the feed system plays havoc with steaming. The boilers steam hard, and foam to such an extent that the coal consumption may easily be doubled. It stands to reason, then, without entering into a technical discussion, that the lower the degree of salinity the greater the evaporative efficiency. The chlorine content is required to be kept below 50 grains of chlorine per gallon of water in large water-tube boilers and 15 grains per gallon in small-tube boilers. This should not be taken as a standard by any means. Every single grain of chlorine has its material effect in reducing a boiler's efficiency, therefore every effort should be exerted toward keeping the chlorine content down to the lowest possible degree. To accomplish this it is best to study first the source of salt in boiler feed.

Salt may enter the feed system—

(1) Through leaky condensers. A tube may split or the ferrules may back off. The condenser may even be improperly packed or may become overheated through a choked main injection.

(2) A distiller may leak either through a split tube or around the tube ends where they are expanded into the tube sheet.

(3) The evaporator coils may leak. This will not admit salt until the evaporators are blown down, when the excess of pressure on the shell over that of the coils, which are then not under pressure, will force concentrated brine into the coil, whence it will make its way through the coil drains into the feed system.

(4) A leak in an evaporator feed-water heater may admit salt if the pressure on the feed water exceeds that on the steam side.

(5) Careless operation of the evaporating plant may admit salt:

(a) If the vapor valves are too wide open and the plant is being forced;

(b) If the distillers are too low in relation to the evaporators;

(c) If the feed water is carried too high;

(d) If the evaporators are not blown down often enough;

(e) If the evaporators are running on the condensers and there is too high a vacuum;

(f) If care is not taken in testing the distilled water;

(6) Salt may enter through sea connections to an auxiliary feed pump;

(7) Through ash ejectors;

(8) Through the feed pump taking a suction from a salty boiler;

(9) Through the carrying over of a highly salty boiler;

(10) Through leaky bottom-blow valves of dead boilers.

To prevent the entrance of salt from these sources the first requisite is an efficient water-testing outfit. This must be capable of determining the grains of chlorine per gallon to at

least one grain. This is required for testing boiler water, but for other tests any solution of silver nitrate will do.

Under (1) Leaky condensers comes first:

(a) Correct method of packing tubes. These should be packed with an approved packing wound in dry or with a little oil. Mixtures of graphite and tallow, or of beeswax and tallow, upon being warmed promptly melt out and leave the ferrules loose. When originally installed the ferrules should be set up hand taut and the condensers filled with water and allowed to soak. The packing, swelling, makes its own joints, which may then be tightened, if necessary.

(b) Every facility should be supplied for quick pressure testing. There should be—

(1) Means for draining and drying the salt-water side;

(2) Hand holes and sighting holes which give access to the tube sheets without removing the heads;

(3) Simple means of filling the fresh-water side with fresh water;

(4) Air connections for admitting compressed air above this water to give constant and uniform pressure;

(5) Valves across main and auxiliary exhaust pipes to isolate the condenser from the machinery;

(6) Accurate pressure gages, and,

(7) Good light on the tube sheets.

(c) There should be a tap on the air-pump discharge from which water may be drawn. This should be tested often by simply dropping a little of the silver nitrate solution in the sample. It must not show the slightest cloud. This degree of tightness can and has been attained. It is paramount in the search for fresh feed.

(2) Distillers should be supplied with connections for quick and easy pressure testing. These are:

(a) Sighting holes through the heads of the tube sheets, and,

(b) Fittings and gages for readily placing a pressure on the steam side. Means should be at hand for obtaining fresh-water samples from each distiller. All fresh water must be

collected in a small tank, and each tankful must be sampled and tested before the water goes to the feed system. To do this it is only necessary to give the evaporator watch a bottle of silver nitrate solution and a medicine dropper. If a few drops of the solution produces a cloud in the sample the tankful should not be sent to the feed system, but to the ship's tank instead.

(3) Leaky evaporator coils may sometimes be detected by taste of the distilled water. To keep them tight, however, it is necessary to have—

(a) Ready means for pressure testing;

(b) A tap on the coil drain by which the drain water may be sampled and tested after blowing down;

(c) A connection for diverting this water to the bilge until it is absolutely fresh as shown by the silver nitrate solution;

(d) Great care and attention that no salt gets by into the feed system by this route.

(4) To prevent leaks past the evaporator feed-water heater are required:

(a) Facilities for quick pressure testing;

(b) A tap on the heater drain for drawing water to be chemically tested, as before.

(5) Careless handling of the evaporators may be prevented by careful supervision and instruction towards keeping—

(a) The evaporator feed water level, low and uniform;

(b) Vapor valves open properly;

(c) Pressures uniform, and,

(d) The evaporators blown down frequently enough to avoid any carrying over.

(6) Sea connections to auxiliary feed pump are to be kept tightly closed.

(7) If salt water from an ash ejector has access to the feed system the connecting piping can be drained before being used for feed line.

(8) No boiler should be used as a feed tank. Shore water usually contains from 1 to 6 grains of chlorine per gallon. To prevent the access of this to the feed line, when used for

make-up feed, the water can first be distilled in the usual way. It evaporates much more quickly than salt water, produces very little wear on the plant and leaves no scale on the coils.

(9) If a very salty boiler carries over it salts up the rest of the battery very rapidly. The boiler can be blown down to reduce chlorine content, or cut out altogether.

Bottom-blow valves demand constant attention and frequent regrinding, especially if the boiler is blown down often.

In distilling water in any way it must be borne in mind that it is not enough that the plant produces "1-grain" or "2-grain" water. Every grain of chlorine promptly finds its way to the boilers and remains there. No feed water is good enough until a 10 per cent. solution of silver nitrate fails to produce even the slightest cloud in a sample. This degree of perfection can and has been attained by locating the salt through the intelligent testing out of each of the sources. When the salt in the feed system is low it is easy to keep the boilers fresh by the usual means. Having a knowledge of the chlorine content of the boiler an engineer may use this information to determine when to cut out the boiler. If it now be filled to the air cock with fresh water the whole boiler is thus freshened and made to steam more easily when next lighted off.

It is readily seen that if economy is to be gained through low chlorine content of the boiler water the evaporating and condensing system must be perfect. There is little economy gained in blowing down a boiler to reduce its degree of salinity, as this results in a loss of fresh water which may offset the gain in evaporative efficiency. It is maintained, however, that if a boiler which is fresh be supplied with absolutely fresh feed it will possess a much higher evaporative efficiency than if it were even slightly salty and supplied with fresh feed. The maximum efficiency results when a fresh boiler is supplied with absolutely fresh feed.

Comparisons of performances on coal consumption involve so many factors that it is extremely difficult to obtain any

valuable information, even when the data is obtained with the utmost care. However, an engineer who keeps in close touch with his plant will observe, if he watches the chlorine content of his boilers carefully, a remarkable relation between the degree of salinity of the boiler water and the cost of operation of the plant. This relation exists even when the chlorine contents varies within such narrow limits as from zero to 20 grains per gallon, as is shown by a few examples.

In a certain coal-burning destroyer, while steaming at 15 knots, two boilers are used and the blowers are run to induce sufficient draft. Other conditions being equal, it has been noted, through a series of observations covering a long period, that half again as great blower speed is required when the chlorine content is 15 grains as when it is 5 grains per gallon. Since the economy varies inversely as the blower speed, which in turn varies directly as the chlorine content, the economy also varies inversely as the chlorine content.

In a certain group of five coal-burning destroyers, observations covering a long period show that if the boats, while steaming in company, be ranked according to the apparent velocity of the smoke-pipe gases at the top of the stacks, the boat apparently steaming the more easily always has the lowest chlorine content, and the standing of the other four are in like proportion, other things being equal.

These two cases are more or less indeterminate owing to the variety of factors in steaming, but one of a number of concrete examples will better establish the point. A certain coal-burning destroyer steaming under the Rules for Engineering Competition, made an average of 298 revolutions of the main engines per minute at full power for two hours. Her boiler water tested 2 grains of chlorine per gallon going out and 3 grains per gallon after the run. Two months later, the plant having been put in even better condition, the boat ran a similar trial and developed but 285 revolutions per minute. The condition, such as the coal used, the draught of the ship, state of the weather and efficiency of personnel, were identical, but the chlorine content of the boilers was 17

grains going out and 19 grains coming back. Later a third trial was held under identical conditions except that all boilers were supplied with 2-grain water. The boat developed 299 revolutions per minute on this trial. This increase her engineer attributes entirely to the fresh boiler water, which tested but 3 grains after the trial. On a boat of this class, where the speed depends entirely upon the evaporative power of the boilers, a marked gain in battle efficiency is thus obtained.

There are numerous other examples. Every engineer, if he take pains to note the chlorine content of his boilers, will find many cases of his own. It will be readily seen that fresh water will result in a distinct gain in the battle efficiency of a vessel in two ways: (1) Increased cruising radius; (2) Increased speed at full power. Just what the increase will be is difficult to determine, but it is obvious that in this day of strict economy, which requires careful analysis to determine the proper percentage of carbon dioxide in the smokepipe gases, the engineer may well devote a portion of his attention to the absolute elimination of salt from the boiler feed water.

TEST OF A WARREN VERTICAL SINGLE BOILER FEED PUMP AT THE NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

The object of the test was to determine the overall mechanical efficiency, water rate in pounds of dry steam per I.H.P. per hour, slip and duty at different piston speeds, discharge pressures and temperatures. The test was conducted under the supervision of Mr. Leo Loeb, M. E.

DESCRIPTION OF THE PUMP.

The pump is of the vertical, single, inside-packed type, and is of the following dimensions:

Number of cylinders	1
Diameter of steam cylinder, inches	9
piston rod, inches	1½
Area of top end of steam piston, square inches	63.62
bottom end of steam piston, square inches	61.85
Nominal stroke, inches	16
Diameter of water piston, inches	6
rod, inches	1½
Area of bottom end of water piston, square inches	28.27
*top end of water piston, square inches	26.50
Number of suction valves	6
Area of each suction valve (net), square inches	2.57
Maximum lift, inch	½
Number of discharge valves	6
Area of each discharge valve (net), square inches	2.57
Maximum lift, inch	½
Working water pressure, pounds per square inch	260
Diameter of steam pipe, inches	1
exhaust pipe, inches	1½
suction pipe, inches	3½
discharge pipe, inches	3

* Water piston packed with four rings of ⅛-inch by ⅛-inch pressed canvas hydraulic packing.

Originally all water valves were of the solid-composition type, but on account of considerable leakage these were replaced by valves of the three-disc type, which proved satisfactory.

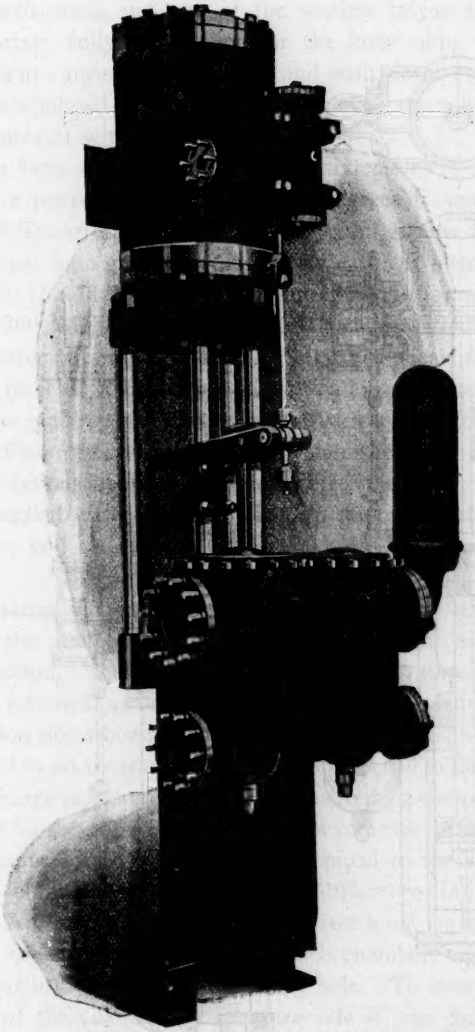
The main valve is of the semi-floating type operated by a pilot valve which controls the motion of the piston, carrying the main valve. The valve gear has one distinct difference from those commonly used on simplex pumps, and that is the fact that the main valve is mechanically operated for a portion of its stroke, roughly 40 per cent., and steam thrown from a point where the pump would otherwise stall. The method of accomplishing this may be seen from section D–E of the cut, showing details. The valve stem terminates in a yoke which envelopes a lug on the main valve and the auxiliary valve A. The valve-rod motion communicated from the crosshead by an oscillating lever carries the main valve and gradually cuts off the steam port. At the same time the auxiliary valve moves and finally throws the main-valve piston and main valve. The yoke is recessed to allow for a motion of the lug of the steam valve for about half the length of the lug. That this action does throttle the steam can be seen from an inspection of the indicator cards, Figs. 1–8.

Two tappets, with tapped bushings, held fast to the valve rod by set screws, shift the valve at the end of the stroke and provide means for adjusting the length of stroke.

The steam piston is cushioned at the end of the stroke by steam admitted through two small auxiliary ports. The main port is not uncovered until the piston has moved about 10 per cent. of its stroke.

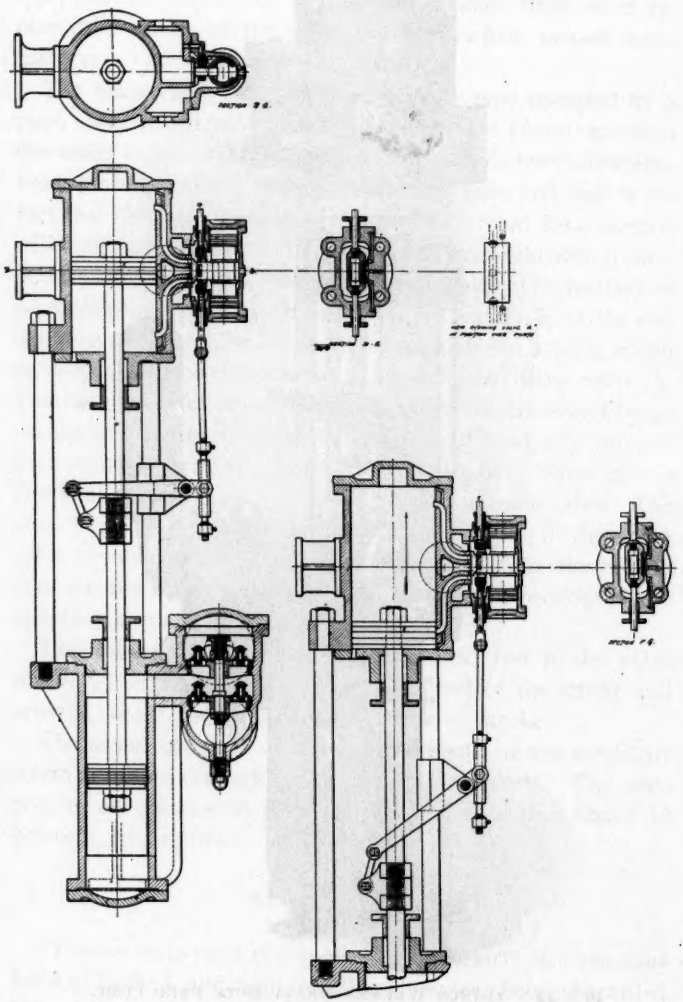
METHOD OF TEST.

Fifty-six tests were run in all, of which thirty-two are tabulated in Table I, and eight in Table II, sixteen being discarded. When the first tests were run for hydraulic efficiency it was found that there was considerable leakage past the valves. The valves were consequently changed from the solid-compo-



16 × 12 × 24-INCH WARREN HEAVY-DUTY FEED PUMP.

sition type to the three-disc type, and an examination of the solid valves which were removed showed that, although the discharge valves seemed to have made fairly tight closure



DETAILS OF WARREN PUMP.

against their seats, not one of the suction valves formed a good bearing, fully accounting for the large slips resulting from tests at varying piston speeds and with closed stop valve. The tests tabulated in Table I were run with the solid valves, and in Table II with the disc valves.

Of the tests tabulated in Table I, sixteen were run with water at a temperature of about 150 degrees F. and sixteen with cold Severn River water. These main groups were subdivided into four groups of four tests each, at piston speeds of 25, 50, 75 and 100 feet per minute and at discharge pressures of 200, 250, 300 and 325 pounds per square inch gage.

The water pumped in cold-water tests was measured by the drop in level of a calibrated 13,000-gallon tank, and at the same time readings were taken on a 3-inch by 1¼-inch Venturi meter. From the calibration thus established and the necessary corrections applied for temperature, the quantity of hot water handled was measured. The duration of each test was one hour, and all observations were taken at ten-minute intervals.

The pump suction was an 8-inch pipe up to within four feet of the pump, where it was reduced to the size of the pump suction. The relative location of suction tank and pump plunger was such as to put at least four-foot pressure head on the suction side when not running. The discharge pressure was measured by an accurate hydraulic gage attached to the center of the discharge pipe, and the suction pressure by a compound gage attached to the suction pipe. For convenience in calculation, the pressure and suction gages were piped to the same level, thereby eliminating correction for difference in height of gages. A 6-inch air chamber, twelve feet long, equalized pressure on the suction side and an 8-inch chamber, eighteen feet long, was attached on the discharge side. To insure the full benefit of the chamber on pressure side it was charged with compressed air to 200 pounds before beginning the tests. Beyond the air chamber an additional pressure gage was installed and after it was determined that there was no appreciable friction drop between it and the gage on the discharge pipe,

readings were taken on the second gage and same averaged with the first.

The I.H.P. was determined from seven sets of indicator cards taken at regular intervals during the test with separate indicators on top and bottom ends of the cylinder. The quality of the steam between throttle valve and steam chest was computed from observations made with a throttling calorimeter. The steam sample was taken across the section of a vertical pipe. A valve was located on the exhaust line between the pump and the condenser by means of which a back pressure of about 5 pounds gage could be maintained on the pump. The exhaust was piped to a condenser, the steam condensed and weighed in tanks on platform scales, which were calibrated and found correct. The condenser was tested for leaks at different times during the test and found tight.

The total travel of the piston was taken by a recording-stroke meter attached to the pump rod, and the number of double strokes by a continuous-rachet counter attached to the pump rod. Additional observations taken were boiler pressures, suction and discharge temperatures, room temperature and steam temperature. Barometer was read once during each test, and a sample of water drawn to determine the density. All thermometers, gages, etc., were calibrated before and after each group of tests and corrections applied where necessary.

RESULTS OF TESTS.

Tables I and II give the average of the data taken and the results as calculated. The results in Table I and corresponding columns in Table II were calculated according to the appendix attached.*

The tests are tabulated according to increasing piston speed at a given discharge pressure, although, on account of simplicity in adjustment, the tests were run according to increasing discharge pressure at a given piston speed.

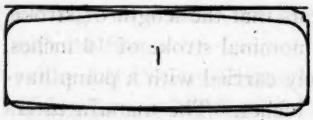
* The data in Table I was collected before the change in valves, and is not correct in regard to the data for hydraulic efficiency. The correct hydraulic efficiency is contained in Table II.

It was noted during the first tests that the length of stroke, while on the average equal to the nominal stroke of 16 inches, was not as great as could be safely carried with a pump having a striking length of over 17 inches. The manufacturers accordingly supplied another auxiliary valve which differed from the first in that the slot which bridges the exhaust ports had a length of $\frac{1}{8}$ inch greater than the first valve, and the overall length of the valve was increased by the same amount. This valve was placed in the pump between tests 16 and 17 of Table I, and its effect on length of stroke is at once noticeable.

Figs. 1 to 8 show specimen cards from both ends of the pump, at various piston speeds, and illustrate the cylinder performance. Cards 1 to 4, inclusive, are from tests with the original auxiliary valve in place, and cards 5 to 8 are with the second auxiliary valve.

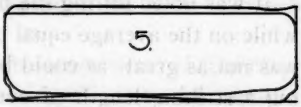
As noted above, all adjustments are made by means of tappets on valve stem. In order to give the pump the benefit of the greatest possible travel the tappets were set so that the pump would not stall at the end of stroke for the lowest discharge pressure at any one piston speed, nor give abnormal cushioning as shown at the exhaust line of indicator cards. No further adjustments were required during the tests at that speed, the higher pressures automatically producing a longer stroke. At the next higher speed the distance between tappets was again increased. The greatest adjustment had to be made between piston speeds of 50 and 75 feet per minute. Between 25 and 50 feet and between 75 and 100 feet the adjustment was small and in some cases could not be disturbed at all.

When first put in service the pump operated at all speeds without noise or vibration. During the 12th test of Table I a slight noise was noted in the water end, increasing slightly with the further service at highest speed and pressure, but proving serious in only three cases, tests 24, 28 and 32, Table I, where the respective indicated strokes of 17.26 inches, 17.39 inches and 17.38 inches are slightly greater than theoretically possible. In those tests the vibration was sufficient



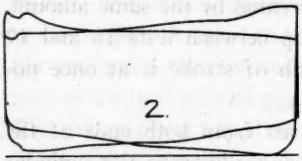
M.E.P. top, 97.1; bottom, 96.9.

Total I.H.P., 4.82.
Spring scale, 100.
Discharge pressure, 199.1.
Piston speed, feet per minute, 26.07.



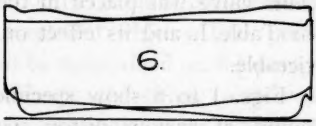
M.E.P. top, 140.9; bottom, 135.0.

Total I.H.P., 6.64.
Spring scale, 100.
Discharge pressure, 301.0.
Piston speed, feet per minute, 25.30.



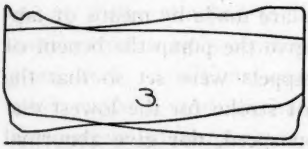
M.E.P. top, 117.8; bottom, 129.3.

Total I.H.P., 11.72.
Spring scale, 100.
Discharge pressure, 249.2.
Piston speed, feet per minute, 49.93.



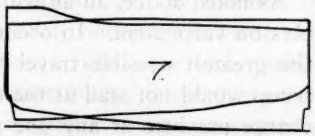
M.E.P. top, 94.4; bottom, 103.9.

Total I.H.P., 9.36.
Spring scale, 100.
Discharge pressure, 201.4.
Piston speed, feet per minute, 49.70.



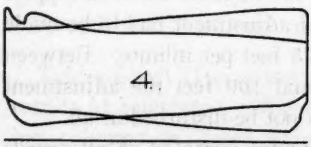
M.E.P. top, 144.6; bottom, 170.3.

Total I.H.P., 22.95.
Spring scale, 150.
Discharge pressure, 323.7.
Piston speed, feet per minute, 76.77.



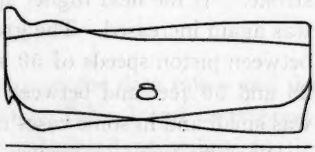
M.E.P. top, 144.0; bottom, 161.4.

Total I.H.P., 21.92.
Spring scale, 150.
Discharge pressure, 325.8.
Piston speed, feet per minute, 75.60.



M.E.P. top, 132.7; bottom, 163.5.

Total I.H.P., 27.89.
Spring scale, 150.
Discharge pressure, 299.0.
Piston speed, feet per minute, 99.20.



M.E.P. top, 136.0; bottom, 156.8.

Total I.H.P., 29.09.
Spring scale, 150.
Discharge pressure, 300.9.
Piston speed, feet per minute, 104.63.

NOTE.—Cards Nos. 1, 2, 3 and 4 were taken from runs made with original auxiliary valve in place; Nos. 5, 6, 7 and 8 from runs with the second auxiliary valve.

CARDS FROM TEST OF 9″ × 6″ × 16″ WARREN FEED PUMP.

to render the data from the stroke meter unreliable, as a decided loop was noticeable in the connecting string between crosshead and meter at each reversal of the pump. In all other tests the action was so smooth as to render the stroke data and all deduced slippage results particularly reliable. After some service the pump developed a slight groan in the steam cylinder at the beginning of top to bottom stroke at slowest speed. Under such conditions the throttling is sufficient to slightly superheat the steam entering the steam chest (note quality in tests, 2, 3, 17 and 18, Table I), and as no lubricant was used, the noise was likely caused by excessive spring-ring pressure against comparatively dry cylinder walls.

Fig. 9 shows the I.H.P. and pounds of steam per I.H.P. per hour at various piston speeds and discharge pressures for cold-water tests. The small numerals indicate the number of the run.

Fig. 10 shows same data for hot-water tests.

Fig. 11 shows the I.H.P., steam in pounds per 1,000 pounds of water pumped, and water discharged in pounds per hour, at various piston speeds and a constant discharge pressure of 250 pounds per square inch gage.

The length of stroke seems to be increased about 0.7 inch by the substitution of the second auxiliary valve. The actual amount varies on either side of this, depending on the tappet adjustment, but an increased length throughout the series is noticeable.

The steam consumption in hot-water tests is still decreasing at 100-foot piston speed, but the rate of decrease is so slow as to indicate its approach to a uniform rate of from 56 pounds to 60 pounds of dry steam per I.H.P. hour. In the cold-water tests the steam consumption is slightly greater than hot-water tests at all speeds, but such increase cannot be charged to the difference in auxiliary valves, as the tendency to produce a longer stroke would lead one to expect better instead of poorer steam consumption. Furthermore, the best steam consumption at the higher discharge pressures is reached at a piston

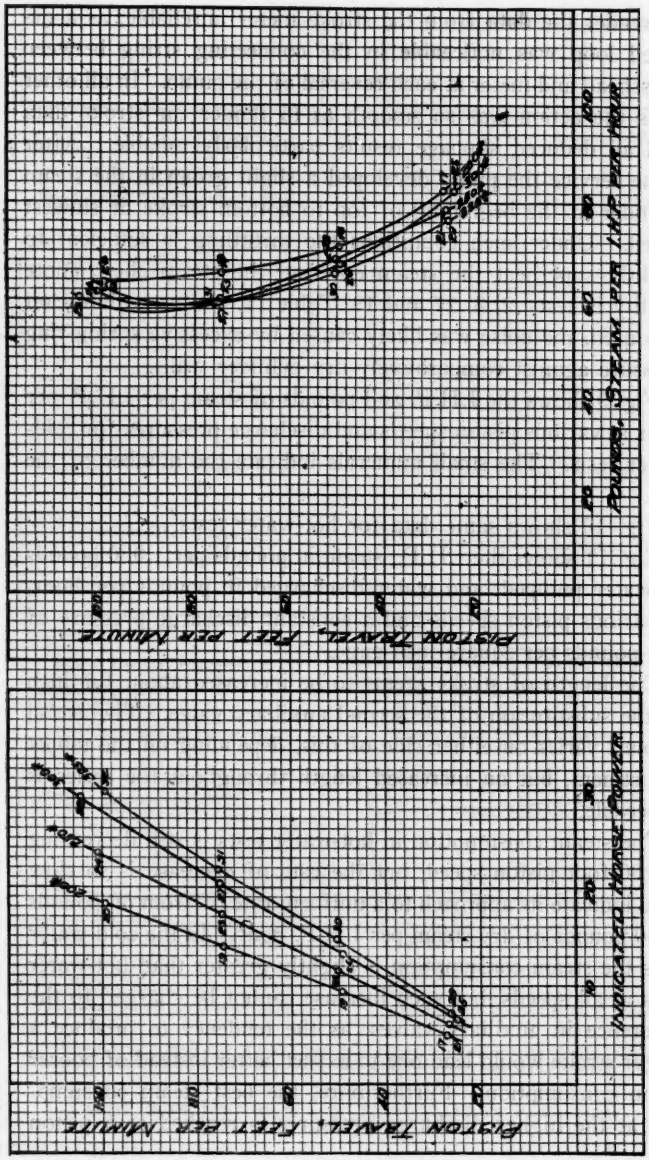


FIG. 9.—CURVES FOR COLD-WATER TESTS OF 9″ × 6″ × 16″ WARREN FEED PUMP.

NOTE.—Small numerals indicate number of run.

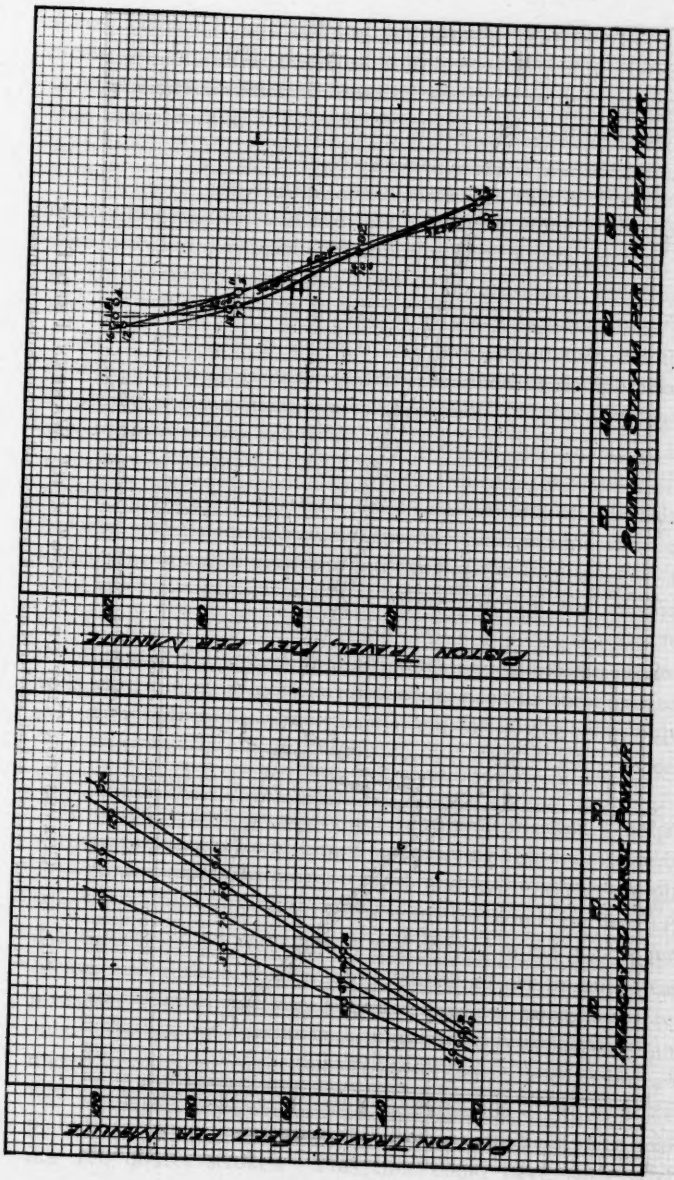

Piston Travel, Feet per Minute
Pounds, Steam per Indicated Horse Power
Piston Travel, Feet per Minute
Indicated Horse Power

FIG. 10.—CURVES FOR HOT-WATER TESTS OF 9" × 6" × 16" WARREN FEED PUMP.

NOTE.—Small numerals indicate number of run.

NOTE.—Small numerals indicate number of run.

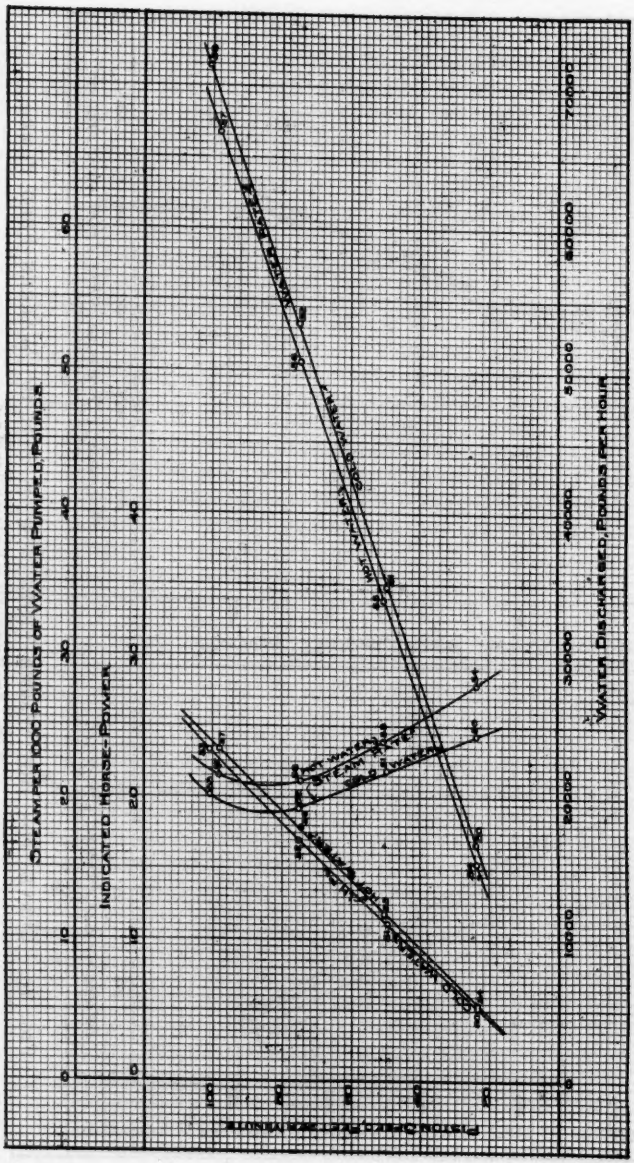

Steam per 1000 Pounds of Water Pumped, Pounds
Indicated Horse-Power
Water Discharged, Pounds per Hour
Piston Speed, Feet per Minute

FIG. 11.—CURVES FOR TEST OF 9″ × 6″ × 16″ WARREN FEED PUMP.

NOTE.—Small numerals indicate number of run. All runs made with 250 pounds discharge pressure.

speed between 75 and 100 feet per minute. The curves of dry steam per thousand pounds of water pumped follow the same general characteristics as those of steam consumption, except that the curvature is more pronounced, due to increasing slip with increasing discharge pressures.

This pump has a good steam economy, due to the partial mechanical operation of the steam valve, producing thereby a wire drawing of steam through the port at the part of the stroke where the requirements of steam are not so great as at reversal. This action is illustrated by the indicator cards Figs. 1 to 8. At a piston speed of 25 feet per minute the total quantity of steam required is such that there is no appreciable throttling, and the steam line is horizontal. At the next higher speed throttling has begun to show itself in that the steam line remains horizontal for about 14 per cent. of the stroke, after which it falls gradually until the end of the stroke. The drop becomes greater at high piston speeds. The peculiar drop and rise in the steam line of the top card at the beginning of the stroke is due to excessive throttling through the auxiliary cushion port. Apparently the port at the top end is restricted, due to faulty drilling of auxiliary port, and the steam pressure falls until the main port is uncovered, rises to chest pressure and then falls again due to partial cut-off by the main valve. The disturbance of the top end becomes greater at high speeds when more steam is required.

In order to test the ability of the pump to start promptly without preliminary warming up, a starting test was made before other machinery was started and with a considerable length of steam pipe filled with water of condensation. All drains were closed and exhaust was opened to the atmosphere. Steam was turned on and observations of water and steam pressure, Venturi meter and strokes made at 30-second intervals. The pump started from near the top of stroke, hammered heavily for two single strokes, then quieted down and was discharging at the rate of 60,000 pounds of water at 300 pounds pressure in four minutes and ten seconds after making 108 double strokes. This time could have with safety

been reduced somewhat.* The only effect on the pump was to start some leakage through the stuffing boxes on steam rod and valve rod owing to the great amount of water in the cylinder.

The pump was easy to start, ran quietly at all speeds, and with a lack of shock at reversal, except in tests 24, 28 and 32.

After changing the valves the pump was put into service and eight tests, against a discharge pressure of 250 pounds gage, were conducted at varying piston speeds and with hot and cold water, as in the first series of tests. Slip tests against closed stop valve were made at three times, each time against discharge pressures of 200 and 325 pounds. In four of the six tests there was some leakage past the stop valve, which leakage was caught, weighed and credited to the pump. The results of the slip tests are shown in Table III.

At highest pressure the slips are at all times in excellent agreement, though there is a considerable variation in slip loss at low pressure. The slip does not appear to have increased with service as the lowest slip at 200 pounds pressure occurred in the final test, whereas the slip at 325 pounds at conclusion of hot-water tests was only 24 pounds per hour in excess of first test. The average slip against closed stop valve for all tests is 259 pounds per hour at 197.1 pounds discharge pressure and 493 pounds per hour at 323.6 pounds discharge pressure. Interpolated to 250 pounds discharge would give 355 pounds per hour. A comparison of this figure with the slips actually obtained shows that with cold water the slip up to and including speeds of 75 feet per minute is not much in excess of that which would be accounted for by slip past plunger and closed suction valves, and the increase above 75 feet piston speed is due to delay in closure of valves.

For cold water the minimum percentage slip is 0.91 per cent. at 75-foot piston speed, whereas the percentage slip is still decreasing at 100-foot piston speed in hot-water tests. However, the actual slip in pounds shows about the same increase

* On a subsequent test, this time was reduced to about one minute.

in each case. It is to be noted that the length of stroke at about 75-foot speed (Table II), cold water, is considerably shorter than other tests, as an adjustment was made necessary by hammer of valves. Short stroking produces a more prompt reversal, and it is not improbable that this may lead to less slip than excessive cushioning on long stroke.

After the change in valves to the disc type, the performance of the pump was satisfactory in every way.

APPENDIX I.

REPORT OF TEST OF 9 INCH × 6 INCH × 16 INCH WARREN BOILER FEED PUMP.

Method of Obtaining Results in Table I.

Item.

1, 2, 3. For reference.

4. From data corrected for temperature according to Smithsonian Meteorological Tables.

5, 6, 7. Observed.

8. Item 7—barometric pressure of item 4 in pounds per square inch.

9. From Data.

10. Item 9—barometric pressure.

11, 12. Average of seven sets of indicator cards.

13. From direct reading stroke meter.

14. $\frac{\text{Item } 13}{60}$.

15. $\frac{(\text{Item } 11 \times A_t + \text{item } 12 \times A_b) \times \text{item } 14}{2 \times 33{,}000}$,

where

A_t = area of top end of cylinder in square inches = 63.62.

A_b = area of bottom end of cylinder in square inches = 61.85.

16. As weighed.

17. Calculated from calorimeter data,

$$x = \frac{H - q}{r}.$$

H = heat content of one pound of steam corresponding to pressure and superheat within calorimeter.

q = heat of the liquid corresponding to item 8.

r = heat of vaporization corresponding to item 8.

In tests 2, 3, 17 and 18 the steam was superheated and its quality in degrees of superheat was calculated from the difference in temperature of a thermometer inserted in steam line and temperature corresponding to item 8.

18. Item 16 × item 17; except tests 2, 3, 17 and 18, where

$$\text{Item 18} = \text{item 16} \times \frac{H^1}{H}.$$

H^1 = total heat of superheated steam corresponding to conditions of item 8 and item 17.

H = total heat of dry and saturated steam corresponding to condition of item 8.

19. Item 18 × total heat of dry saturated steam corresponding to pressure in item 8.

20. Assuming that the quality of exhaust steam is 0.85, and the steam is used in a feed-water heater of 90 per cent. efficiency, then this column would be obtained as follows:

$(.85\ r+q) \times$ item 18 $\times 0.9$

where

r = heat of vaporization corresponding to item 10.

q = heat of the liquid corresponding to item 10.

21. Item 19—item 20.

22. Assuming that the condensed steam from the pump is available as boiler feed at 200 degrees F., then the total B.t.u. absorbed by the pump per hour will be:

Item 18 $\times (r+q-q^1)$,

where

r = heat of vaporization corresponding to item 8.

q_1 = heat of liquid corresponding to item 8.

q = heat of the liquid at 200 degrees F. = 167.9.

23. From data.

24. From data.

25. Item 23—item 24.

26. From data.

27. From chemist's determination.

28. Item 25 $\times \frac{144}{\text{Item 27}}$.

29. Drop in level of calibrated tank correct for density, item 27, for tests 17 to 32, inclusive, and Venturi meter observations corrected for temperature, item 26, in tests 1 to 16, inclusive.

30. Item 13 × item 27 × .1902.

where .1902 is the cubic feet displaced by water plunger for each foot of travel.

31. Item 30—item 29.

32. From data.

33. $\frac{H \times W}{60 \times 33{,}000}$,

where

H = total discharge head in feet = item 28.

W = water delivered per hour in pounds = item 29.

TABLE I.—TEST OF 9" × 6" × 16" WARREN FEED PUM

					Data from Steam End.																	
Test Number.	Date — October, November, 1912.	Duration of Test. Minutes.	Barometer. Corrected. Inches of Hg.	Room Temperature. Degrees Fahr.	Steam Pressure at Boiler. Pounds Gage.	Steam Pressure in Chest. Pounds Gage.	Steam Pressure in Chest. Pounds Absolute.	Back Pressure. Pounds Gage.	Back Pressure. Pounds Absolute.	M.E.P. of Steam Cylinder. Top End.	M.E.P. of Steam Cylinder. Bottom End.	Piston Travel. Feet per Hour.	Piston Speed. Feet per Minute.	I.H.P. Developed by Steam Cylinder.	Steam Consumed per Hour. Pounds.	Quality of Steam.	Dry Steam Consumed per Hour. Pounds.	Total B.T.U. Supplied to Pump per Hour.	B.T.U. per Hour Rejected and Used in Heating Feed Water.	Heat Consumed by Pump. B.T.U. per Hour.	Heat Consumed by Pump, According to Test, Less Friction. B.T.U. per Hour.	Pressure in Main on Delivery Side. Pounds Gage.	Pressure on Suction
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	
1	29	60	29.97	78.5	310.0	107.5	122.2	4.95	19.67	97.1	96.9	1564	26.07	4.82	410.0	.9666	396.3	471438	360831	110607	404900	198.6	
2	30	60	29.96	79.5	310.0	123.9	138.6	5.30	20.01	96.3	106.8	2950	49.17	9.48	710.0	6.8°s	713.2	850206	649654	200552	730459	198.4	
3	30	60	30.13	74.4	312.3	130.9	145.7	4.76	19.55	96.2	110.7	4484	74.73	14.68	922.0	9.9°s	928.5	1107515	845306	262209	951620	196.8	
4	31	60	30.03	74.7	308.3	142.7	157.5	4.67	19.41	92.6	111.9	6042	100.70	19.55	1184.0	.9917	1174.2	1402934	1068874	334060	1205786	197.9	
5	29	60	29.94	77.1	310.0	132.5	147.2	5.84	20.54	120.5	120.8	1500	25.00	5.73	486.5	.9831	478.3	570899	435970	134929	490592	248.7	
6	30	60	29.91	81.2	310.0	148.4	163.1	5.17	19.86	117.8	129.3	2996	49.93	11.72	849.0	.9978	847.1	1012962	771451	241508	870734	248.2	
7	31	60	30.12	75.0	312.6	159.5	174.2	4.81	19.60	116.7	134.7	4508	75.13	17.93	1093.5	.9940	1087.1	1300389	989696	310693	1117865	247.1	
8	1	60	29.77	73.7	310.3	177.1	191.8	4.81	19.43	109.4	141.7	6052	100.87	24.03	1417.5	.9735	1379.9	1652154	1256261	395893	1420469	246.9	
9	29	50	29.94	76.2	310.0	156.4	171.1	5.14	19.84	139.1	146.5	1454	24.23	6.58	558.0	.9900	552.4	660670	503236	157434	567922	258.7	
10	30	60	29.91	83.7	310.0	181.1	195.8	5.32	20.01	139.4	153.0	2996	49.93	13.87	1017.0	.9829	999.6	1197221	910536	286685	1029388	259.1	
11	31	60	30.09	75.6	311.3	199.0	213.8	4.94	19.71	135.1	158.5	4494	74.90	20.87	1350.0	.9734	1314.1	1575606	1196488	379118	1354969	258.0	
12	1	60	29.62	74.1	311.6	206.7	221.2	5.15	19.69	132.7	163.5	5952	99.20	27.89	1631.0	.9508	1550.7	1859910	1412067	447843	1599547	257.8	
13	31	60	29.96	78.5	310.0	172.9	187.7	4.95	19.66	157.6	161.4	1400	23.33	7.07	570.0	.9979	568.8	681138	517892	163246	585636	323.1	
14	30	60	29.90	83.5	310.0	193.0	207.7	5.41	20.09	149.7	161.7	2974	49.57	14.66	1082.5	.9767	1057.3	1267386	963095	304291	1089865	322.8	
15	31	60	30.07	75.7	310.1	206.6	221.3	4.92	19.68	144.6	170.3	4606	76.77	22.95	1410.5	.9716	1370.4	1643384	1247749	395535	1413294	322.5	
16	1	60	29.59	75.1	308.6	221.0	235.5	5.75	20.28	142.2	173.6	6086	101.43	30.40	1772.0	.9502	1683.7	2020897	1534447	486450	1738084	322.6	
17	21	60	30.12	72.7	275.4	108.1	122.9	4.80	19.59	95.9	98.0	1640	27.33	5.04	414.5	2.1°s	414.8	493778	377634	116144	424133	202.4	
18	23	60	29.88	71.7	291.6	120.6	135.3	4.75	19.43	94.4	103.9	2982	49.70	9.36	664.5	3.5°s	665.6	793661	605896	187765	681907	202.4	
19	23	60	29.72	75.4	300.0	133.9	148.5	4.82	19.42	92.2	105.1	4470	74.50	13.96	915.5	.9997	915.2	1092566	833015	259551	938904	201.3	
20	25	60	29.88	60.1	303.6	142.9	157.6	5.03	19.71	85.6	108.5	5964	99.40	18.31	1195.0	.9725	1162.1	1387664	1058092	329572	1192547	199.0	
21	21	60	30.10	74.1	275.0	132.6	147.4	4.80	19.58	118.5	119.8	1612	26.87	6.09	482.0	.9931	478.7	571137	435808	135329	490763	251.1	
22	23	60	29.82	73.3	300.7	146.6	161.3	4.80	19.45	116.7	125.8	3014	50.23	11.57	791.5	.9991	790.8	944848	719865	224983	812073	250.5	
23	25	60	29.83	57.6	303.0	156.2	170.9	4.80	19.46	118.8	124.1	4482	74.70	17.24	1047.0	.9893	1035.8	1239024	942889	296135	1065113	249.8	
24	25	60	29.91	61.9	299.0	183.0	197.7	4.91	19.61	107.1	138.9	6054	100.90	23.55	1521.5	.9523	1448.9	1735203	1319223	415980	1491932	249.2	
25	21	60	30.08	75.3	275.3	155.9	170.7	5.18	19.95	140.9	135.0	1518	25.30	6.64	556.0	.9829	546.5	653286	497807	155479	561529	303.3	
26	23	60	29.78	74.1	299.1	170.8	185.4	4.62	19.25	136.4	146.9	2970	49.50	13.32	904.5	.9923	897.5	1074128	816815	257313	923438	301.9	
27	25	60	29.81	60.1	301.4	188.7	203.4	4.53	19.18	135.0	149.9	4504	75.07	20.31	1260.0	.9551	1203.4	1441794	1095094	346700	1239743	300.9	
28	26	60	30.05	65.3	300.7	226.1	240.9	6.00	20.76	136.0	156.8	6278	104.63	29.09	1849.5	.9511	1759.1	2111096	1603947	507149	1815743	300.3	
29	21	60	30.04	75.0	274.7	163.2	178.0	4.66	19.42	150.9	137.7	1570	26.17	7.18	551.0	.9968	549.2	656953	499882	157071	564742	327.5	
30	23	60	29.73	75.3	299.3	186.7	201.3	4.65	19.26	148.2	158.2	3022	50.37	14.67	971.0	.9885	959.8	1149936	873514	276422	988786	325.8	
31	25	60	29.82	61.7	300.7	199.9	214.6	4.67	19.32	144.0	161.4	4536	75.60	21.92	1367.0	.9516	1300.8	1559529	1183988	375541	1341125	326.1	
32	26	60	29.96	71.0	308.6	239.8	254.5	6.16	20.88	146.3	170.0	5958	99.30	29.82	1948.5	.9501	1851.3	2223967	1688200	535767	1913133	326.0	

FEED PUMP.—TABLE OF DATA AND RESULTS.

Pump Lowering Feed to Heater to 200°F. B.T.U. per Hour.	Data from Water End.											Results.								
	Pressure in Main on Delivery Side, Pounds Gage.	Pressure on Suction Side, Pounds Gage.	Total Pressure on Pump, Pounds.	Temperature of Suction, Degrees Fahr.	Density of Water, Pounds per Cubic Foot.	Total Discharge Head in Feet.	Water Delivered per Hour, Pounds.	Theoretical Piston Displacement, Pounds per Hour.	Slip, Pounds per Hour.	Number of Double Strokes per Hour.	Water Horsepower Developed.	Duty Calculated from Column 21, Foot Pounds.	Duty Calculated from Column 22, Foot Pounds.	Percentage of Slip.	Theoretical Capacity, Gallons per Hour.	Actual Capacity, Gallons per Hour.	Length of Stroke in Inches.	Overall Mechanical Efficiency.	Dry Steam per I.H.P. per Hour, Pounds.	Dry Steam per 1000 Pounds of Water
22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42
24900	198.6	-0.5	199.1	136.3	61.87	463.4	14413	18406	3993	607	3.37	603848 20	16495390	21.70	2226	1743	15.46	69.9	82.2	27.9
30459	198.4	-1.0	199.4	149.3	61.62	466.0	31100	34575	3475	1118	7.32	722263950	19840400	10.05	4198	3776	15.86	77.2	75.2	22.93
31620	196.8	-1.0	197.8	147.2	61.66	461.9	48171	52990	4419	1656	11.24	848566680	23381380	8.40	6381	5844	16.28	76.6	63.2	19.27
05786	197.9	-1.0	198.9	141.0	61.78	463.6	61880	71004	9124	2183	14.49	858755500	23791590	12.85	8998	7493	16.61	74.1	60.1	18.98
90592	248.7	-0.5	249.2	142.9	61.74	581.2	13882	17614	3732	582	4.07	597960 30	16445880	21.19	2134	1682	15.46	71.0	83.5	34.46
70734	248.2	-1.0	249.2	149.2	61.62	582.1	31217	35117	3900	1137	9.18	752414 70	20869080	11.11	4263	3790	15.82	78.3	72.3	27.14
17865	247.1	-1.0	248.1	148.8	61.60	580.0	47863	52822	4959	1672	14.02	893503 90	24833535	9.39	6415	5813	16.12	78.2	60.6	22.71
20469	246.9	-1.2	248.1	139.5	61.80	578.1	61030	71144	10114	2203	17.82	891186 30	24837880	14.22	8612	7388	16.48	74.2	57.4	22.61
67922	298.7	-0.5	299.2	142.7	61.75	697.7	14525	17080	2555	568	5.12	643704 20	17644163	14.95	2069	1760	15.36	77.8	84.0	36.03
29388	299.1	-1.0	300.1	149.2	61.62	701.3	31547	35117	3570	1136	11.17	771715 00	21492300	10.17	4263	3830	15.82	80.5	72.1	31.69
54969	298.0	-1.0	299.0	151.3	61.58	699.2	47772	52639	4867	1635	16.87	881049 80	24651620	9.25	6395	5804	16.49	80.8	63.0	27.51
99547	297.8	-1.2	299.0	147.8	61.64	698.5	60067	69783	9716	2190	21.19	933686 00	26230430	13.92	8469	7290	16.31	76.0	55.6	25.82
85636	323.1	-0.5	323.6	146.4	61.67	755.8	13996	16423	2427	548	5.34	647818 50	18057940	14.78	1992	1698	15.36	75.5	80.5	40.64
89865	322.8	-1.0	323.8	150.4	61.59	757.0	31168	34841	3673	1132	11.92	775382 00	21648710	10.54	4232	3786	15.74	81.3	72.1	33.92
13294	322.5	-1.2	323.7	150.3	61.60	756.7	46902	53968	7066	1695	17.92	897284 50	25112070	13.09	6554	5696	16.33	78.1	59.7	29.22
38084	322.6	-1.2	323.8	149.8	61.61	756.8	60633	71320	10687	2239	23.18	943304 60	26400940	14.98	8660	7362	16.31	76.3	55.4	27.77
24133	202.4	1.1	201.3	55.2	62.87	461.1	14130	19609	5479	611	3.29	560971 12	15361556	27.94	2333	1681	16.10	65.3	82.3	39.36
81907	202.4	1.0	201.4	55.0	62.88	461.2	29748	35666	5918	1094	6.93	730688 78	20119720	16.59	4243	3539	16.36	74.0	71.1	22.37
38904	201.3	0.5	200.8	52.8	62.89	459.8	45967	53469	7502	1594	10.67	814314 99	22510999	14.03	6360	5468	16.82	76.4	65.6	19.91
92547	199.0	-0.5	199.5	49.0	62.92	456.6	62422	71376	8954	2100	14.39	864815 12	23930010	12.54	8486	7422	17.04	78.6	63.5	18.62
90763	251.1	0.2	250.9	55.8	62.87	574.7	12822	19276	6454	600	3.72	544510 27	15014993	33.48	2294	1526	16.13	61.1	78.6	37.33
12073	250.5	-0.1	250.6	55.4	62.88	573.9	28451	36049	7598	1105	8.25	725745 01	20106603	21.08	4289	3385	16.37	71.3	68.3	27.79
65113	249.8	0.2	249.6	50.1	62.92	571.2	45347	53639	8292	1638	13.08	874675 33	24318740	15.46	6378	5392	16.42	75.9	60.1	22.84
91932	249.2	-0.3	249.5	49.0	62.92	571.0	61825	72452	10627	2105	17.83	848648 37	23661987	14.67	8614	7351	17.26	75.7	61.5	23.44
61529	303.3	2.0	301.0	54.7	62.87	690.1	11969	18151	6182	563	4.17	531249 04	14709493	34.06	2160	1424	16.19	62.8	82.3	45.66
23438	301.9	1.1	300.8	53.2	62.89	688.7	28496	35527	7031	1092	9.91	762697 38	21252315	19.79	4226	3390	16.32	74.4	67.4	31.50
39743	300.9	0.1	300.8	49.0	62.92	688.4	46052	53904	7852	1636	16.01	914398 52	25571588	14.57	6409	5475	16.52	78.8	59.3	26.13
15743	300.3	-0.6	300.9	50.1	62.94	688.4	63084	75157	12073	2165	21.93	856297 18	23916945	16.06	8933	7498	17.39	75.4	60.5	27.89
64742	327.5	1.4	326.1	57.0	62.87	746.9	11238	18773	7535	582	4.24	534386 49	14862826	40.14	2234	1337	16.20	59.1	76.5	48.87
88786	325.8	0.0	325.8	53.3	62.89	746.0	27746	36149	8403	1111	10.45	748801 33	20933262	23.25	4300	3301	16.33	71.2	65.4	34.99
41125	326.1	0.3	325.8	49.2	62.92	745.6	45462	54282	8819	1648	17.12	902603 63	25274652	16.25	6454	5405	16.51	78.1	59.3	28.61
13133	326.0	-0.4	326.4	49.8	62.94	746.8	59753	71324	11571	2057	22.54	893689 79	23324850	16.22	8477	7102	17.38	75.6	62.1	30.96

TABLE I.—TEST OF 9" × 6" × 16" WARREN FEED PUMP.—TA

Test Number.	Date — October, November, 1912.	Duration of Test. Minutes.	Barometer. Corrected. Inches of Hg.	Room Temperature. Degrees Fahr.	Data from Steam End.																		Pressure in Main on Delivery Side. Pounds Gage.	Pressure on Suction Side. Pounds Gage.	Total Pressure on Pump. Pounds.
					Steam Pressure at Boiler. Pounds Gage.	Steam Pressure in Chest. Pounds Gage.	Steam Pressure in Chest. Pounds Absolute.	Back Pressure. Pounds Gage.	Back Pressure. Pounds Absolute.	M.E.P. of Steam Cylinder. Top End.	M.E.P. of Steam Cylinder. Bottom End.	Piston Travel. Feet per Hour.	Piston Speed. Feet per Minute.	I.H.P. Developed by Steam Cylinder.	Steam Consumed per Hour. Pounds.	Quality of Steam.	Dry Steam Consumed per Hour. Pounds.	Total B.T.U. Supplied Pump per Hour.	B.T.U. per Hour Rejected or Used in Heating Feed Water.	Heat Consumed by Pump. B.T.U. per Hour.	Heat Consumed by Pump, Reduced to Heat Equiv. to 1 B.T.U. per Hour.			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
1	29	60	29.97	78.5	310.0	107.5	122.2	4.95	19.67	97.1	96.9	1564	26.07	4.82	410.0	.9666	396.3	471438	360831	110607	404900	198.6	-0.5	199.
2	30	60	29.96	79.5	310.0	123.9	138.6	5.30	20.01	96.3	106.8	2950	49.17	9.48	710.0	6.8°s	713.2	850206	649654	200552	730459	198.4	-1.0	199.
3	30	60	30.13	74.4	312.3	130.9	145.7	4.76	19.55	96.2	110.7	4484	74.73	14.68	922.9	9.9°s	928.5	1107515	845306	262209	951620	196.8	-1.0	197.
4	31	60	30.03	74.7	308.3	142.7	157.5	4.67	19.41	92.6	111.9	6042	100.70	19.55	1184.0	.9917	1174.2	1402934	1068874	334060	1205786	197.9	-1.0	198.
5	29	60	29.94	77.1	310.0	132.5	147.2	5.84	20.54	120.5	120.8	1500	25.00	5.73	486.5	.9831	478.3	570899	435970	134929	490592	248.7	-0.5	249.
6	30	60	29.91	81.2	310.0	148.4	163.1	5.17	19.86	117.8	129.3	2996	49.93	11.72	849.0	.9978	847.1	1012962	771454	241508	870734	248.2	-1.0	249.
7	31	60	30.12	75.0	312.6	159.5	174.2	4.81	19.60	116.7	134.7	4508	75.13	17.93	1093.5	.9940	1087.1	1300389	989696	310693	1117865	247.1	-1.0	248.
8	1	60	29.77	73.7	310.3	177.1	191.8	4.81	19.43	109.4	141.7	6052	100.87	24.03	1417.5	.9735	1379.9	1652154	1256261	395893	1420469	246.9	-1.2	248.
9	29	50	29.94	76.2	310.0	156.4	171.1	5.14	19.84	139.1	146.5	1454	24.23	6.58	558.0	.9900	552.4	660670	503236	157434	567922	298.7	-0.5	299.
10	30	60	29.91	83.7	310.0	181.1	195.8	5.32	20.01	139.4	153.0	2996	49.93	13.87	1017.0	.9839	999.6	1197221	910536	286685	1029388	299.1	-1.0	300.
11	31	60	30.09	75.6	311.3	199.0	213.8	4.94	19.71	135.1	158.5	4494	74.90	20.87	1350.0	.9734	1314.1	1575606	1196488	379118	1354969	298.0	-1.0	299.
12	1	60	29.62	74.1	311.6	206.7	221.2	5.15	19.69	132.7	163.5	5952	99.20	27.89	1631.0	.9508	1550.7	1859910	1412067	447843	1599547	297.8	-1.2	299.
13	31	60	29.96	78.5	310.0	172.9	187.7	4.95	19.66	137.6	161.4	1400	23.33	7.07	570.0	.9979	568.8	681138	577892	163246	585636	323.1	-0.5	323.
14	30	60	29.90	83.5	310.0	193.0	207.7	5.41	20.09	149.7	161.7	2974	49.57	14.66	1082.5	.9767	1057.3	1267386	963095	304291	1089865	322.8	-1.0	323.
15	31	60	30.07	75.7	310.1	206.6	221.3	4.92	19.68	144.6	170.3	4606	76.77	22.95	1410.5	.9716	1370.4	1643384	1247749	395535	1413294	322.5	-1.2	323.
16	1	60	29.59	75.1	308.6	221.0	235.5	5.75	20.28	142.2	173.6	6086	101.43	30.40	1772.0	.9502	1683.7	2020897	1534447	486450	1738084	322.6	-1.2	323.
17	21	60	30.12	72.7	275.4	108.1	122.9	4.80	19.59	95.9	98.0	1640	27.33	5.04	414.5	2.1°s	414.8	493778	377634	116144	424133	202.4	1.1	201.
18	23	60	29.88	71.7	291.6	120.6	135.3	4.75	19.43	94.4	103.9	2982	49.70	9.36	664.5	3.5°s	665.6	793661	605896	187765	681907	202.4	1.0	201.
19	23	60	29.72	75.4	300.0	133.9	148.5	4.82	19.42	92.2	105.1	4470	74.50	13.96	915.5	.9997	915.2	1092566	833015	259551	938904	201.3	0.5	200.
20	25	60	29.88	60.1	303.6	142.9	157.6	5.03	19.71	85.6	108.5	5964	99.40	18.31	1195.0	.9725	1162.1	1387664	1058092	329572	1192547	199.0	-0.5	199.
21	21	60	30.10	74.1	275.0	132.6	147.4	4.80	19.58	118.5	119.8	1612	26.87	6.09	482.0	.9931	478.7	571137	435808	135329	490763	251.1	0.2	250.
22	23	60	29.82	73.3	300.7	146.6	161.3	4.80	19.45	116.7	125.8	3014	50.23	11.57	791.5	.9991	790.8	944848	719865	224983	812073	250.5	-0.1	250.
23	25	60	29.83	57.6	303.0	156.2	170.9	4.80	19.46	118.8	124.1	4482	74.70	17.24	1047.0	.9893	1035.8	1239024	942889	296135	1065113	249.8	0.2	249.
24	25	60	29.91	61.9	299.0	183.0	197.7	4.91	19.61	107.1	138.9	6054	100.90	23.55	1521.5	.9523	1448.9	1735203	1319223	415980	1491932	249.2	-0.3	249.
25	21	60	30.08	75.3	275.3	155.9	170.7	5.18	19.95	140.9	135.0	1518	25.30	6.64	556.0	.9829	546.5	653286	497807	155479	561529	303.3	2.0	301.
26	23	60	29.78	74.1	299.1	170.8	185.4	4.62	19.25	136.4	146.9	2970	49.50	13.32	904.5	.9983	897.5	1074128	816815	257313	923438	301.9	1.1	300.
27	25	60	29.81	60.1	301.4	188.7	203.4	4.53	19.18	135.0	149.9	4504	75.07	20.31	1260.0	.9551	1203.4	1441794	1095094	346700	1239743	300.9	0.1	300.
28	26	60	30.05	65.3	300.7	226.1	240.9	6.00	20.76	136.0	156.8	6278	104.63	29.09	1849.5	.9511	1759.1	2111096	1603947	507149	1815743	300.3	-0.6	300.
29	21	60	30.04	75.0	274.7	163.2	178.0	4.66	19.42	150.9	137.7	1570	26.17	7.18	551.0	.9968	549.2	656953	499882	157071	564742	327.5	1.4	326.
30	23	60	29.73	75.3	299.3	186.7	201.3	4.65	19.26	148.2	158.2	3022	50.37	14.67	971.0	.9885	959.8	1149936	873514	276422	988786	325.8	0.0	325.
31	25	60	29.82	61.7	300.7	199.9	214.6	4.67	19.32	144.0	161.4	4536	75.60	21.92	1367.0	.9516	1300.8	1559529	1183988	375541	1341125	326.1	0.3	325.
32	26	60	29.96	71.0	308.6	239.8	254.5	6.16	20.88	146.3	170.0	5958	99.30	29.82	1948.5	.9501	1851.3	2223967	1688200	535767	1913133	326.0	-0.4	326.

PUMP.—TABLE OF DATA AND RESULTS.

Data from Water End.											Results.								
on Delivery Side. Pounds Gage.	Pressure on Suction Side. Pounds Gage.	Total Pressure on Pump. Pounds.	Temperature of Suction. Degrees Fahr.	Density of Water. Pounds per Cubic Foot.	Total Discharge Head in Feet.	Water Delivered per Hour. Pounds.	Theoretical Piston Displacement. Pounds per Hour.	Slip. Pounds per Hour.	Number of Double Strokes per Hour.	Water Horsepower Developed.	Duty Calculated from Column 21. Foot Pounds.	Duty Calculated from Column 22. Foot Pounds.	Percentage of Slip.	Theoretical Capacity. Gallons per Hour.	Actual Capacity. Gallons per Hour.	Length of Stroke in Inches.	Overall Mechanical Efficiency.	Dry Steam per I.H.P. per Hour, Pounds.	Dry Steam per 1000 Pounds of Water Pumped, Pounds.
23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42
8.6	-0.5	199.1	136.3	61.87	463.4	14413	18406	3993	607	3.37	60384820	16495390	21.70	2226	1743	15.46	69.9	82.2	27.80
8.4	-1.0	199.4	149.3	61.62	466.0	31100	34575	3475	1118	7.32	72263550	19840400	10.05	4198	3776	15.86	77.2	75.2	22.93
6.8	-1.0	197.8	147.2	61.66	461.9	48171	52590	4419	1656	11.24	84856680	23381380	8.40	6381	5844	16.28	76.6	63.2	19.27
7.9	-1.0	198.9	141.0	61.78	463.6	61880	71004	9124	2183	14.49	85875500	23791590	12.85	8998	7493	16.61	74.1	60.1	18.98
8.7	-0.5	249.2	142.9	61.74	581.2	13882	17614	3732	582	4.07	59796030	16445880	21.19	2134	1682	15.46	71.0	83.5	34.46
8.2	-1.0	249.2	149.2	61.62	582.1	31217	35117	3900	1137	9.18	75241470	20869080	11.11	4263	3790	15.82	78.3	72.3	27.14
7.1	-1.0	248.1	149.8	61.60	580.0	47863	52822	4959	1672	14.02	89350390	24833535	9.39	6415	5813	16.12	78.2	60.6	22.71
6.9	-1.2	248.1	139.5	61.80	578.1	61030	71144	10114	2203	17.82	89118630	24837880	14.22	8612	7388	16.48	74.2	57.4	22.61
8.7	-0.5	299.2	142.7	61.75	697.7	14525	17080	2555	568	5.12	64370420	17844163	14.95	2069	1760	15.36	77.8	84.0	38.03
9.1	-1.0	300.1	149.2	61.62	701.3	31547	35117	3570	1136	11.17	77171500	21492300	10.17	4263	3830	15.82	80.5	72.1	31.69
8.0	-1.0	299.0	151.3	61.58	699.2	47772	52639	4867	1635	16.87	88104980	24651620	9.25	6395	5804	16.49	80.8	63.0	27.51
7.8	-1.2	299.0	147.8	61.64	698.5	60067	69783	9716	2190	21.19	93686400	26230430	13.92	8469	7290	16.31	76.0	55.6	25.82
3.1	-0.5	323.6	146.4	61.67	755.8	13996	16423	2427	548	5.34	64781850	18057940	14.78	1992	1698	15.36	75.5	80.5	40.64
2.8	-1.0	323.8	150.4	61.59	757.0	31168	34841	3673	1132	11.92	77538200	21648710	10.54	4232	3786	15.74	81.3	72.1	33.92
2.5	-1.2	323.7	150.3	61.60	756.7	46902	53968	7066	1695	17.92	89728450	25112070	13.09	6554	5696	16.33	78.1	59.7	29.22
2.6	-1.2	323.8	149.8	61.61	756.8	60633	71320	10687	2239	23.18	94330460	26400940	14.98	8660	7362	16.31	76.3	55.4	27.77
2.4	1.1	201.3	55.2	62.87	461.1	14130	19609	5479	611	3.29	56097112	15361556	27.94	2333	1681	16.10	65.3	82.3	39.36
2.4	1.0	201.4	55.0	62.88	461.2	29748	35666	5918	1094	6.93	73068878	20119720	16.59	4243	3539	16.36	74.0	71.1	22.37
1.3	0.5	200.8	52.8	62.89	459.8	45967	53469	7502	1594	10.67	81431499	22510999	14.03	6360	5468	16.82	76.4	65.6	19.91
9.0	-0.5	199.5	49.0	62.92	456.6	62422	71376	8954	2100	14.39	86481512	23930010	12.54	8486	7422	17.04	78.6	63.5	18.62
1.1	0.2	250.9	55.8	62.87	574.7	12822	19276	6454	600	3.72	54451027	15014993	33.48	2294	1526	16.13	61.1	78.6	37.33
0.5	-0.1	250.6	55.4	62.88	573.9	28451	36049	7598	1105	8.25	72574501	20106603	21.08	4289	3385	16.37	71.3	68.3	27.79
9.8	0.2	249.6	50.1	62.92	571.2	45347	53639	8292	1638	13.08	87467533	24318740	15.46	6378	5392	16.42	75.9	60.1	22.84
9.2	-0.3	249.5	49.0	62.92	571.0	61825	72452	10627	2105	17.83	84864837	23661987	14.67	8614	7351	17.26	75.7	61.5	23.44
3.3	2.0	301.0	54.7	62.87	690.1	11969	18151	6182	563	4.17	53124904	14709493	34.06	2160	1424	16.19	62.8	82.3	45.66
1.9	1.1	300.8	53.2	62.89	688.7	28496	35527	7031	1092	9.91	76269738	21252315	19.79	4226	3390	16.32	74.4	67.4	31.50
0.9	0.1	300.8	49.0	62.92	688.4	46052	53904	7852	1636	16.01	91439852	25571588	14.57	6409	5475	16.52	78.8	59.3	26.13
0.3	-0.6	300.9	50.1	62.94	688.4	63084	75157	12073	2165	21.93	85629718	23916945	16.06	8933	7498	17.39	75.4	60.5	27.89
7.5	1.4	326.1	57.0	62.87	746.9	11238	18773	7535	582	4.24	53438649	14862826	40.14	2234	1337	16.20	59.1	76.5	48.87
5.8	0.0	325.8	53.3	62.89	746.0	27746	36149	8403	1111	10.45	74880133	20933262	23.25	4300	3301	16.33	71.2	65.4	34.59
6.1	0.3	325.8	49.2	62.92	745.6	45462	54281	8819	1648	17.12	90260363	25274652	16.25	6454	5405	16.51	78.1	59.3	28.61
6.0	-0.4	326.4	49.8	62.94	746.8	59753	71324	11571	2057	22.54	83289079	23324850	16.22	8477	7102	17.38	75.6	62.1	30.98

					Data from Steam End.								
Test Number.	Date. Jan..1913.	Duration of Test, Minutes.	Barometer Corrected, Inches of Mercury.	Room Temperature, Degrees Fahr.	Steam Pressure in Chest, Pounds Gage.	Steam Pressure in Chest, Pounds Absolute.	Back Pressure, Inches of Mercury Gage.	Back Pressure, Inches of Mercury, Absolute.	Piston Travel, Feet per Hour.	Piston Speed, Feet per Minute.	I.H.P. Developed by Steam Cylinder.	Dry Steam Consumed per Hour, Pounds.	Total B.T.U. Supplied
1	2	3	4	5	6	7	8	9	10	11	12	13	
33	15	45	30.54	64.0	133.7	148.7	23.43	7.11	1435	23.92	5.16	405.6	4
34	15	45	30.52	64.0	152.5	167.5	20.75	9.77	2976	49.60	11.12	759.5	9
35	15	45	30.48	67.0	161.8	176.8	18.50	11.98	4491	74.85	16.98	1020.5	12
36	15	45	30.46	67.0	186.0	201.0	15.70	14.76	6056	100.93	23.34	1435.4	17
37	16	30	30.27	69.0	127.4	142.3	24.83	5.44	1432	23.87	5.44	427.6	5
38	16	30	30.25	68.5	148.1	163.0	20.86	9.39	3028	50.47	11.84	808.7	9
39	16	30	30.24	69.0	167.0	181.9	19.71	10.53	4472	74.53	17.86	1073.4	12
40	16	30	30.25	69.0	181.9	196.8	16.29	13.96	5880	98.00	23.28	1431.7	17

TABLE II.—TEST OF 9" × 6" × 16" WARREN FEED PUMP.—TABLE OF DATA

					Date. from Wate					
per Hour. Pounds.	Total B.T.U. Supplied Pump per Hour.	B.T.U. per Hour Rejected and Used in Heating Feed Water.	Heat Consumed by Pump, B.T.U. per Hour.	Heat Consumed by Pump Assuming Feed Heated to 200° Fahr.. B.T.U. per Hour.	Pressure in Main on Delivery Side. Pounds Gage.	Pressure on Suction Side. Pounds Gage.	Total Pressure on Pump. Pounds.	Temperature of Suction. Degrees Fahr.	Density of Water. Pounds per Cubic Foot.	Total Discharge Head. in Feet.
13	14	15	16	17	18	19	20	21	22	23
405.6	484327	369258	115069	415821	247.7	1.6	246.1	50.0	62.91	563.
759.5	906159	691373	214786	779931	251.4	0.6	250.8	50.0	62.91	574.
020.5	1219293	928961	290332	1049380	250.6	0.7	249.9	43.0	62.92	572.
435.4	1717025	1306932	410093	1478051	252.8	0.5	252.3	41.0	62.92	577.
427.6	510597	389287	121310	438376	251.1	0.1	251.0	146.6	61.62	586.
808.7	964860	736160	228700	830454	250.8	-0.1	250.9	149.9	61.55	587.
073.4	1282498	977116	305382	1103777	251.6	-0.2	251.8	148.3	61.58	588.
431.7	1712600	1303563	409037	1474221	251.6	-0.5	252.1	147.4	61.60	589.

× 6″ × 16″ Warren Feed Pump.—Table of Data and Results.

	Data from Water End.											
Heat Consumed by Pump Assuming Feed to be Heated to 200° Fahr., B.T.U. per Hour.	Pressure in Main on Delivery Side, Pounds Gage.	Pressure on Suction Side, Pounds Gage.	Total Pressure on Pump, Pounds.	Temperature of Suction, Degrees Fahr.	Density of Water, Pounds per Cubic Foot.	Total Discharge Head, in Feet.	Water Delivered per Hour, Pounds.	Theoretical Pump Discharge from Plunger Displacement, Pounds per Hour.	Slip, Pounds per Hour.	Number of Double Strokes per Hour.	Water Horsepower Developed.	Duty Calculated from Col. 16, Foot Pounds.
17	18	19	20	21	22	23	24	25	26	27	28	29
415821	247.7	1.6	246.1	150.0	62.91	563.3	16672	17171	499	515.3	4.74	81614840
779931	251.4	0.6	250.8	150.0	62.91	574.1	34824	35610	786	1074.7	10.10	93080824
1049380	250.6	0.7	249.9	143.0	62.92	572.0	53256	53746	490	1658.7	15.40	104922753
1478031	252.8	0.5	252.3	141.0	62.92	577.5	71417	72474	1057	2170.7	20.83	100570644
438376	251.1	0.1	251.0	146.6	61.62	586.6	15328	16783	1455	511.0	4.54	74119244
830454	250.8	-0.1	250.9	149.9	61.55	587.1	33784	35448	1664	1081.0	10.02	86727536
1103777	251.6	-0.2	251.8	148.3	61.58	588.7	50602	52378	1776	1611.0	15.05	97547977
1474221	251.6	-0.5	252.1	147.4	61.60	589.4	66689	68892	2203	2092.0	19.85	96095211

Results.								
Duty Calculated from Col.16, Foot Pounds.	Duty Calculated from Col.17, Foot Pounds.	Percentage of Slip.	Theoretical Capacity, Gallons per Hour.	Actual Capacity, Gallons per Hour.	Length of Stroke. Inches.	Overall Mechanical Efficiency.	Dry Steam per I.H.P. per Hour, Pounds.	Dry Steam per 1000 Pounds of Water Pumped, Pounds.
29	30	31	32	33	34	35	36	37
81614840	22585050	2.91	2042	1982	16.70	91.8	78.6	24.33
93080824	25633624	2.21	4235	4140	16.60	90.8	68.3	21.81
04922751	29028981	0.91	6390	6330	16.24	90.7	60.1	19.16
00570646	27904231	1.46	8617	8490	16.73	89.2	61.5	20.10
74119240	20510715	8.67	2038	1861	16.82	83.5	78.6	27.90
86727530	23884027	4.69	4309	4106	16.80	84.7	68.3	23.94
97547979	26988601	3.39	6363	6147	16.75	84.2	60.1	21.21
96095211	26662554	3.20	8367	8099	16.88	85.2	61.5	21.47

34. $\frac{H \times W \times 1{,}000{,}000}{h}$,

where

H = total discharge head in feet = item 28.

W = water delivered per hour in pounds = item 29.

h = total heat units consumed by pump per hour = item 21.

35. As in item 34, except that h = total heat consumed = item 22.

36. $\frac{\text{Item } 31}{\text{Item } 30}$.

37. Item 13 × .1902 × 7.481,

where

.1902 = cubic feet of plunger displacement per foot of travel,

7.481 gallons = 1 cubic foot.

38. Item 37 × $\frac{100 - \text{item } 36}{100}$.

39. $\frac{\text{Item } 13 \times 12}{2 \times \text{item } 32}$.

40. $\frac{\text{Item } 33}{\text{Item } 15}$.

41. $\frac{\text{Item } 18}{\text{Item } 15}$.

42. $\frac{\text{Item } 18 \times 1{,}000}{\text{Item } 29}$.

TABLE III.—SLIP TESTS.

Test Number.	Date, January, 1913.	Following Test No.	Duration, minutes.	Double strokes.	Double strokes, per hour.	Length of stroke, inches.	Piston displacement, pounds per hour.	Leakage past stop valve, pounds per hour.	Net slip, pounds per hour.	Water pressure, pounds gage.
1	15	32	12.05	2	9.96	16.7	332	95	237	194.6
2	15	32	6.22	2	19.32	16.7	644	171	473	325.0
3	15	36	5.48	2	23.62	16.7	786	405	381	197.6
4	15	36	5.17	2	25.10	16.7	836	325	511	319.5
5	15	40	25.13	2	4.78	16.9	158	0	158	199.1
6	15	40	7.97	2	15.04	16.9	497	0	497	326.4

ALASKAN NAVAL RADIO EXPEDITION, 1912.

By Lieutenant E. H. Dodd, U. S. Navy, Member.

During the summer of 1911 an expedition from the Mare Island Navy Yard established temporary radio stations at Kodiak, Dutch Harbor and St. Paul, Pribilof Islands, Alaska, and selected Unalga Island as the best location in the vicinity of Unimak Pass for a medium-power radio station.

The expedition of 1912 was organized for the purposes of erecting the Unalga station, making the installations at Dutch Harbor and St. Paul permanent, installing a small station at St. George, Pribilof Islands, rebuilding the Kodiak station (which had been destroyed by fire during the eruption of Mt. Katmai), installing new apparatus in the Cordova station and overhauling and making minor changes in the station at Sitka.

The U. S. S. *Nero* was assigned by the Navy Department for transporting the personnel and material of the expedition. The equipment and material was assembled at Mare Island, and necessarily had to be complete in every detail, and in anticipation of emergencies and accidents. Practically no material could be obtained at the different places visited, and, with the exception of Cordova and Sitka, the limited time allowed for the expedition would have prevented obtaining any material by shipment from the United States.

About four months was required for assembling and stowing material on the *Nero* and fitting the vessel for the cruise. The *Nero* carried 1,010 tons of material for the expedition and about 40 tons of provisions, exclusive of live stock.

The preparation costs of the expedition were practically 13 per cent. of the total cost, and included working-party equipment, bedding, mess outfits, tools, machines, engines, motors, supplies and labor in handling same.

The personnel of the expedition, in addition to the complement of the collier, consisted of Lieutenant E. H. Dodd, U. S. Navy, Mare Island radio officer; Surgeon G. W. R. French, U. S. Navy; Electrical Expert Aide Geo. E. Hanscom; thirty-one navy-yard workmen, and thirty-eight enlisted men of the Navy, almost all of whom were petty officers. Not all of the enlisted men were with the expedition at any one time, because of transfers to and from the different stations visited.

The itinerary of the *Nero*, W. J. Kelton, N. A. S., Master, was as follows:

1912.

May 20. Left Mare Island and arrived San Francisco.

21. Left San Francisco.

Towing U. S. S. *Fox* and *Davis*.

24. Off Tatoosh Island, Washington.

Let go of *Fox* and *Davis*.

29. Hove to for 20 hours in storm.

June 2. Arrived at Dutch Harbor, Alaska.

7. Left Dutch Harbor and arrived at Unalga Island, Alaska.

While at anchor off Unalga, from the date of above arrival until the final departure from that station on September 6, 1912, on six occasions the *Nero* was forced to run to Dutch Harbor for shelter or put to sea and heave to on account of violent gales.

15. Left Unalga and arrived at Dutch Harbor.

Working party of 45 men remained at Unalga.

30. Left Dutch Harbor and arrived at Unalga.

Working party of 4 men remained at Dutch Harbor.

Aug. 1. Left Unalga.

Working party of 46 men remained at Unalga. Vessel stopped at Dutch Harbor and picked up four men who were left there July 30th.

Aug. 2. Hove to for 8 hours in storm.
3. Arrived at St. George Island, Pribilof Ids., Alaska.
7. Left St. George and arrived St. Paul Island, Pribilof Ids.

While off St. Paul the vessel was frequently forced to shift anchorage and twice to put to sea and heave to on account of severe storms.

26. Left St. Paul.

Two men of working party remained at St. Paul and left there Sept. 5, 1912, on the U. S. R. C. *Tahoma*, arriving at Unalaska Sept. 7, 1912.

29. Arrived Unalga.

Vessel was delayed by storm and stopped at Dutch Harbor to await favorable tide for going through Unalga Pass.

Sept. 6. Left Unalga and arrived at Dutch Harbor.
8. Left Dutch Harbor.

Departure was delayed 12 hours by storm.

12–13. Hove to 12 hours in storm.
14. Arrived Cordova, Alaska.
Oct. 10. Left Cordova.
12. Arrived Kodiak, Alaska.
Nov. 5. Left Kodiak.

Departure was delayed 36 hours by storm.

8. Arrived Sitka, Alaska.
13. Left Sitka.

Departure was delayed one day awaiting mail steamer. For five days after leaving strong southeast gales and heavy seas were encountered.

22. Arrived San Francisco.
23. Left San Francisco and arrived Mare Island.

Total time of cruise, 6 months, 3 days.

The weather conditions of the whole cruise were trying. Storms were the rule and fair weather was the exception. Every trip was delayed by storms, except two—from Cordova to Kodiak, and Kodiak to Sitka. At Unalga the expedition experienced good weather for 26 per cent. of the time, and

storms, with a wind velocity of 75 miles an hour or over, for 27 per cent. of the time. On days not included above there was either heavy rain or strong winds and rain. The first wharf built by the expedition was washed away on June 19th. Succeeding storms wrecked two lighters, and the last storm completely wrecked the pontoon used in landing stores. Outside work was not possible during the storms.

Bad weather was also encountered at Dutch Harbor and Pribilof Islands. At the latter station communication between ship and shore was difficult at all times and often impossible. There were only four working days at St. Paul without wind and rain. There were two days without rain at Cordova, and there was 26 inches of rain during the first two weeks at that station. Weather at Kodiak was fair, but on some days the temperatnre was below freezing. The wind at times blew great clouds of ashes down from the mountain side, making it difficult to recognize objects 50 yards away. During the time the expedition was at Kodiak there were a number of intermittent earthquake shocks. Good weather prevailed during the five days the expedition was at Sitka.

The Alaska stations are in isolated locations, and, outside of the radio work, the men detailed to them lead a lonely existence, with little opportunity for amusement or association with other persons. Provisions are difficult to obtain; mail is infrequent. The steamer *Dora* stops once a month at Dutch Harbor and also at Unalga on the same trip, if the occasion arises and weather permits. From the first part of November to about the first of June no vessels stop at Pribilof. The island supply ship makes two trips each summer from San Francisco, and revenue cutters, which base during the summer at Dutch Harbor, call at different times when on Behring Sea patrol duty. A steamer calls at Kodiak once in three weeks, and steamers reach Cordova and Sitka once each week. However, all operators assigned to Alaskan stations volunteered for the duty. There are five operators and a cook at Unalga, two at Dutch Harbor, two at Pribilof and four operators and a cook at Cordova, Kodiak and Sitka.

One of the principal objects of the Alaskan stations is to provide radio communication for isolated districts. All communicate with shipping, furnish weather reports on request and handle commercial business. During the summer the stations at St. Paul and St. George assist the revenue cutters in protecting the Pribilof Islands from seal poachers. During winter and spring these stations afford the only communication between those islands and the outside world. Dutch Harbor also works in conjunction with the patrol fleet, the headquarters of which is in Unalaska. The radio route is also used in case of a break in the Seattle-Valdez cable or the land telegraph line from Valdez to Nome. The cable broke just before the expedition reached Cordova, and while the work could have been handled by Sitka and Cordova new installation at the latter station would have been held up. Consequently, cable business was routed between Sitka and Nome via St. Paul, Pribilof. On a number of occasions the Sitka and St. Paul stations would be working on wave lengths of 1,000 and 1,400 meters, respectively, at the same time as Unalga and Pacific coast stations were working on 2,000 meters, and no station interfered with any of the others, indicating that all were sharply tuned.

ALASKAN NAVAL RADIO COMMUNICATIONS.

1. Unalga (Akutan Pass, Alaska).

Ten-kw. quenched-spark, high-frequency set. Wave lengths: 1,600 to 3,500 meters. Uses 1,800 meters and above for long-distance communication, 1,800 for long distance with ships, and will "listen in" on that tune for 15 minutes following 11:00 P. M. and midnight.

Three-kw. auxiliary set, rotary spark. Wave lengths: 300, 600 and 1,000 meters for local work with ships and nearby stations.

Storage-battery set, at times, for communication within radius of 60 miles.

Handles commercial business. Constant watch. Long-

distance work with other stations usually between midnight and 4:00 A. M., station time.

Communicates night with San Diego, San Francisco, Mare Island, Eureka, North Head, Sitka, Cordova, Kodiak, Alaskan cannery stations, Nome and Honolulu.

Day and night with Dutch Harbor, Pribilof, and probably later on with North Head, Cordova, Naknek, Nome and Honolulu.

Also communicates with revenue cutters, steamers crossing the North Pacific, those on the Nome-Seattle route, and whaling vessels at Akutan.

2. Dutch Harbor (Alaska).

Five-kw., quenched-spark, high-frequency set. Wave lengths: 300, 600 and 1,000 to 3,800 meters. Uses 1,400 meters and above for long-distance communication; 300, 500 and 1,000 for local work with ships and nearby stations.

Storage-battery set, at times, for communication within radius of 40 miles.

Handles commercial business. Following is watch schedule: Listens in and works, if called, for 15 minutes following the hours at 10:00 and 11:00 A. M., and 4:00, 5:00, 6:00, 7:00 and 8:00 P. M., and continuously from 9:00 P. M. until 1:00 A. M., or as soon after 1:00 A. M. as business is cleared. Will send messages at any hour, day or night.

Communicates night with North Head, Sitka, Cordova, Kodiak, Alaska cannery stations, Nome, and has communicated with Mare Island and Eureka.

Day and night with Unalga, Pribilof and Naknek.

Also communicates with revenue cutters, vessels crossing the North Pacific, those on the Nome-Seattle route, and whaling vessels at Akutan.

3. Pribilof (St. Paul, Pribilof Ids., Alaska).

Five-kw., quenched-spark, high-frequency set. Wave lengths: 300, 600 and 1,000 meters to 4,200 meters. Uses 1,400 meters and above for long-distance communication;

300, 600 and 1,000 for local work with ships and nearby stations.

Storage-battery set, at times, for communication within radius of 125 miles.

Handles commercial business. Following is watch schedule: Listens in and works, if called, for 15 minutes following the hours at 10:00 and 11:00 A. M., and 4:00, 5:00, 6:00, 7:00 and 8:00 P. M., and continuously from 9:00 P. M. until 1:00 A. M., or as soon after 1:00 A. M. as business is cleared. Will send messages at any hour, day or night. Long-distance work usually 11:00 P. M. and 1:00 A. M., station time.

Communicates night with Sitka, Cordova, Kodiak, Alaska cannery stations, and frequently with Mare Island, Eureka and North Head.

Day and night with St. George, Unalga, Dutch Harbor, Naknek and Nome.

Also communicates with revenue cutters, vessels crossing the North Pacific and on the Nome-Seattle route.

4. St. George (Pribilof Islands, Alaska).

Auxiliary station to St. Paul, Pribilof; not manned by naval operator, but will be during the summer months.

One-half-kw. set and auxiliary storage-battery set. Wave length: 300 meters.

Handles commercial business, but communicates only with St. Paul, Pribilof, station and ships in the immediate vicinity. No schedule of watches, but clears daily with St. Paul.

5. Cordova (Alaska).

Ten-kw., quenched-spark, high-frequency set. Wave lengths: 300, 600, 1,000 and 1,500 to 3,300 meters. Uses 1,500 meters and above for long-distance communication; 1,800 for long-distance with ships, and will listen in on that tune for 15 minutes following 11:00 P. M. and midnight; 300, 600 and 1,000 for local work with ships and nearby stations.

Storage-battery set, at times, for communication within radius of 50 miles.

Handles commercial business. Constant watch. Long-distance work usually from midnight to 4:00 A. M., station time.

Communicates night with San Diego, San Francisco, Mare Island, Eureka, North Head, Tatoosh, Alaska cannery stations, Unalga, Dutch Harbor, Pribilof and Honolulu.

Day and night with Sitka and Kodiak, and probably later on with North Head and Unalga.

Also communicates with local shipping, with steamers on Seattle-Cordova-Valdez-Seward route, on Nome-Seattle route and those crossing the North Pacific.

Has cable connections to Alaska military cable office at Cordova.

Kodiak (Alaska; Woody Island).

Five-kw., quenched-spark, high-frequency set. Wave lengths: 300, 600 and 1,400 to 6,000 meters. Uses 1,400 and above for long-distance communication; 1,800 for long-distance with ships; 300, 600 and 1,000 for local work with ships and nearby stations.

Storage-battery set, at times, for communication within a radius of 50 miles.

Also storage-battery auxiliary station in Kodiak village for communicating with main station.

Handles commercial business. Constant watch. (Only the auxiliary station listens in from 9:00 to 11:00 A. M., and 1:00 to 3:00 P. M., and 5:00 to 7:00 P. M., and clears with the main station immediately before and after each of the above periods.)

Long-distance work usually from 11:00 P. M. to 2:00 A. M., station time.

Communicates night with North Head, Tatoosh, Alaska cannery stations, Unalga, Dutch Harbor, Pribilof, Honolulu, Mare Island and Eureka.

Day and night with Cordova, Sitka, and frequently with Alaska cannery stations and Afognak.

Also communicates with vessels crossing the North Pacific,

on Nome-Seattle route, Valdez-Seward, on Seattle-Cordova route and with local shipping.

Sitka (Alaska).

Ten-kw. set, rotary spark. Wave lengths: 300, 600 and 1,000 to 2,000 meters. Uses 1,000 meters and above for long-distance communication; 1,800 for long distance with ships; 300, 600 and 1,000 for local work with ships and nearby stations.

Storage-battery set, at times, for communication within radius of 30 miles.

Handles commercial business. Constant watch. Long-distance work usually from 10:00 P. M. to 2.00 A. M., station time.

Communicates night with Mare Island, Eureka, North Head, Tatoosh, British Columbian stations, Ketchikan, St. Petersburg, Wrangle, Juneau, cannery stations in vicinity, Unalga, Dutch Harbor, Pribilof and Honolulu.

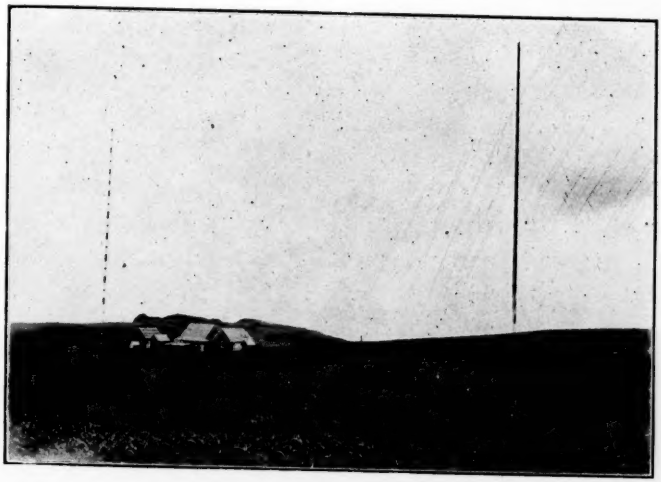
Day and night with Cordova and Kodiak.

Also communicates with local shipping, steamers crossing the North Pacific, and on southwestern and southeastern Alaskan routes.

Has cable connection to Alaska military cable office at Sitka.

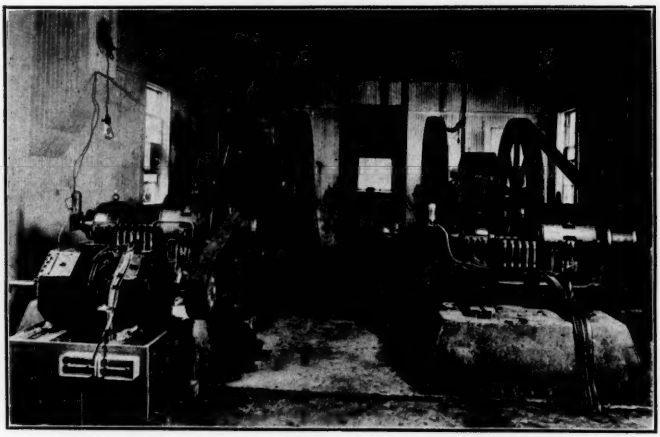
UNALGA.

Unalga was an uninhabited island between the Akutan and Unalga Passes, about 25 miles from Dutch Harbor. The expedition arrived June 7th, and the material and equipment for the station was landed in 23 days, which were not consecutive, due to weather conditions. It was often necessary to continue operations until late at night to take advantage of favorable tides and good weather. A wharf 180 feet long was built in 5 days and utilized until June 19th, when it was carried away during a storm. The wreckage was cleared away on June 22 and the construction of a new wharf was begun, as it was impossible to land material by any other

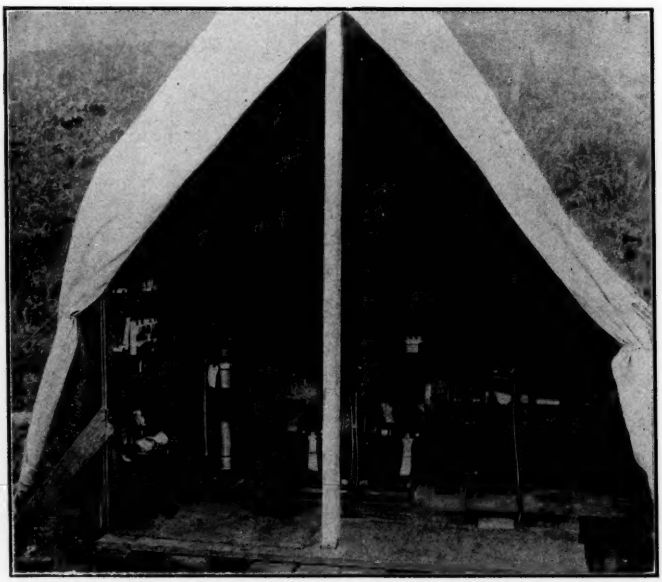


UNALGA STATION.—REAR VIEW.

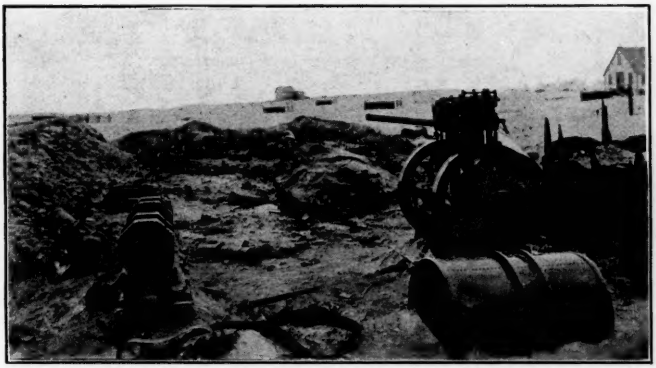
UNALGA.—DINING ROOM.



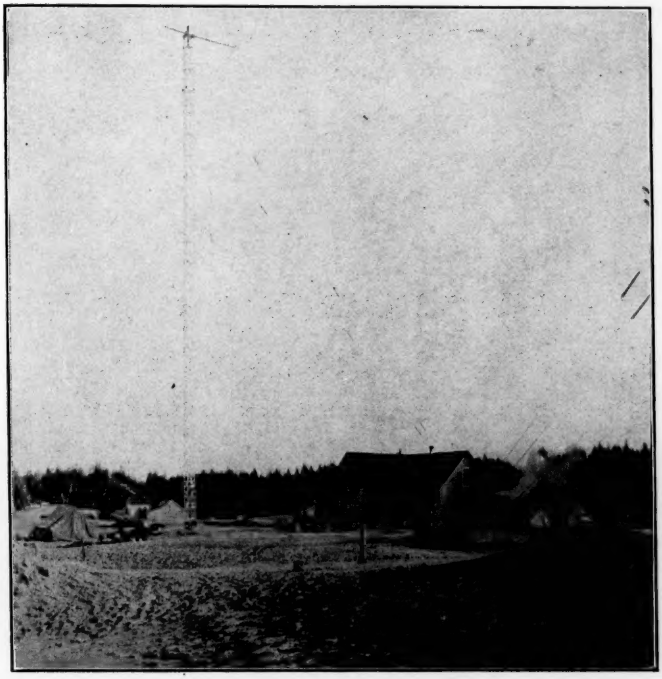
CORDOVA.——25-H.P. KEROSENE ENGINES. 10-KW. ALTERNATERS FOR MAIN SET, AND 3-KW. ALTERNATER FOR CHARGING STORAGE BATTERIES AND ELECTRIC LIGHTING.



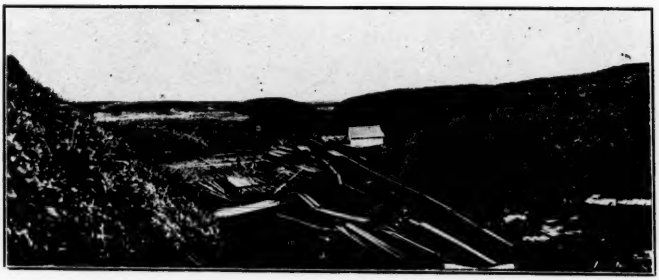
UNALGA. —— POWER PLANT FOR TEMPORARY RADIO SET, CHARGING STORAGE BATTERIES AND LIGHTING QUARTERS OF WORKING PARTY.



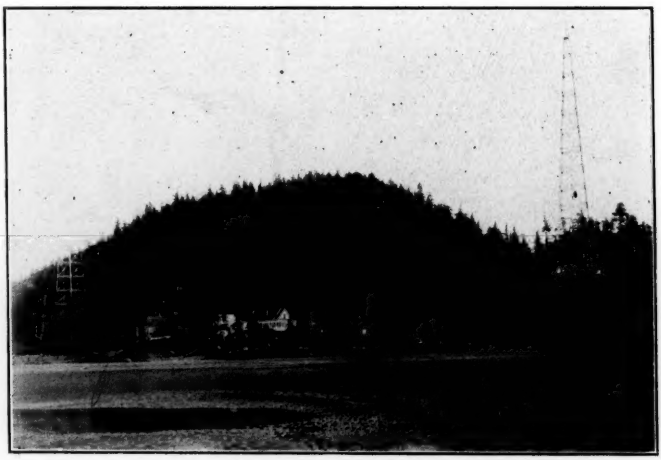
RUINS OF KODIAK STATION WHICH WAS STRUCK BY LIGHTNING JUNE 8, 1912, DURING THE ERUPTION OF MT. KATMAI.



KODIAK.—COMPLETED NEW STATION.

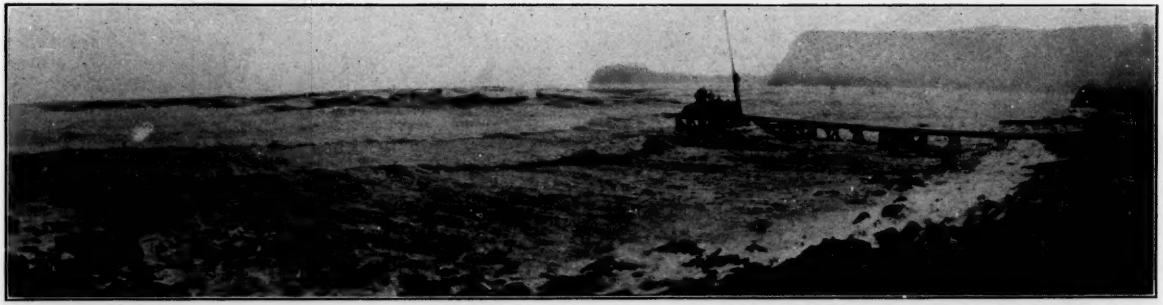


UNALGA. — LOCATION OF WORKING PARTY CAMP. STARTING BUNK HOUSE. (X) SITE OF RADIO STATION.



CORDOVA STATION.

LANDING MATERIAL AT UNALGA. THE WHARF WAS CARRIED AWAY AND THE LARGE BARGE TOTALLY WRECKED DURING STORM OF JUNE 19, 1912.



COMMODORE COVE, UNALGA, DURING STORM OF SEPTEMBER 5, 1912.

means without entailing great delay. The wharf was completed in 6 days, a derrick rigged at its outer end and an industrial railroad installed on it. The railroad was extended to the working-party camp and a 5-H.P. stationary gas engine was used for hauling the cars up the 4 per cent. grade. The railroad was later extended to the station site and a 20-H.P. donkey engine installed for hauling the cars up the grade of 18.6 per cent. Considerable cribbing was necessary to cross swampy tundra and small surface streams. There were two cars, each of two tons capacity. Another railroad was built along the shore line from the shore end of the wharf to a gravel beach 1,050 feet distant. The two small cars for this road were moved by man power, and there was a slight down grade for the loaded cars.

Material was landed as fast as possible and stowed at various stations between the wharf and the camp. Tents were set up as protection for some of the material, and other material was covered with canvas. It was necessary to build a shed for the cement, and the inside was dried for three days with ten oil heaters before the cement could be put in it. As soon as the railroad was extended to the station site material was moved to that point and sorted and segregated on arrival.

The bunk house for the working party consisted of a dormitory 50 feet by 20 feet, dining room 38 feet by 20 feet, and kitchen, store-rooms and wash-room. In the dormitory were 58 bunks arranged in tiers of three. The officers' quarters were in a tent near the bunk house.

When the working party first landed a temporary storage-battery radio set was installed in a tent near the site of the station for communicating with the *Nero* and Dutch Harbor. On the 26th, the experimental radio set of the working party was installed in the tent, and later moved to the oil house. The set consisted of an 8-jar condenser, induction coil, break key and 1-P-76 receiving set. The generating apparatus, set up in a tent near the bunk house, consisted of a 6-H.P. gas engine, driving a 1-kw. self-excited 100-cycle alternator. As

the exciting coils of the alternator were burnt out, a 5-H.P., D.C., motor, brought with the expedition for running a saw bench, was used for exciting the fields and was driven with a leather belt from the same engine. Power leads were run to the tent in which the set was installed. Owing to danger of fire from oil lamps in bunk house, electric lighting was installed and current taken from the 5-H.P. exciter of the temporary radio set. This exciter, with a rheostat, was also used for charging storage batteries. A temporary telephone service was installed with four stations at important locations.

Unalga Island is composed mainly of volcanic rock covered with moss and tundra, but on the level where the station was located soil was found. In digging for anchors and foundations strata were as follows:

Tundra or moss, inches,	8
Sand, inches,	5
Brown clay, inches,	10
Black loam, inches,	10
Cinders, inches,	7
Yellow clay, inches,	12
Yellow clay and gravel,	8
Yellow clay and boulders,	30
Rock,	

The actual construction work at the station required about two months. The other time was required in handling material, temporary construction work, surveying, testing and adjusting, etc. A saw table was set up on the site, roofed for protection against rain, and driven by the 10-H.P. gas engine later used for the auxiliary radio set. A power-driven concrete mixer was of greatest assistance in saving both time and labor. It was necessary to build a dam in a small stream near the station, and run a pipe line, with gravity flow, to tanks set up temporarily, from which pipe and hose were led to the various engine and mast foundations and anchors. The station lathe was utilized as soon as set up. The work-

ing-party motor-driven lathe had been installed on the *Nero* and was also used when required.

A tent was set up for glazing windows and another as a machinist's tool and workshop. The oil house, in addition to its use for the temporary radio set, was utilized for calibrating, testing and adjusting, and measuring instruments and equipment of radio set. The coal house, which was the first building erected, was used as a combined tool-shed, workshop and drafting room. Weather interfered with the work, but advantage was taken of one good day to put up the power-house flooring, studding, rafters and over half of the outside sheathing.

The masts were laid off in the direction of Mare Island and Eureka. All material for their construction, except cement, was purchased in San Francisco for $7,490. The concrete foundations were 5 feet by 5 feet by 2 feet, each weighing about two tons, and the anchors 7 feet by 4 feet by 5 feet top and 7 feet bottom, each weighing about 14 tons. Each mast was assembled on the ground, in ten sections of 30 feet each, and hoisted into place by the donkey engine. An 80-foot gin pole was used, with necessary fittings for shifting it up the mast as sections were landed and rigging set up. A six-foot 4 × 4 extension was placed on each mast for the single-wire aerial. The first section of the east mast was installed July 26th, and the last section after five working days, *i. e.*, days when weather permitted work on the mast. Better time was made with the west mast, which took only four days. The donkey engine was skidded along by its own power from the first to second mast, and after completion of the latter was skidded back again for railroad work. The masts were in no way affected by the high wind during the severe storm of September 5th.

The buildings at Unalga consist of a dwelling, in which is located the operating room, a power house with a covered passage to the dwelling, and a cottage for the electrician in charge. The buildings have shingled roofs and sides and are fitted with storm windows throughout. A glass porch extends

all around the dwelling and is of particular value in offering protection against cold winds and rain, and muffling outside noise which interferes with receiving. The inside of each building is finished with plaster board and battens. All buildings have electric lights and the dwelling and cottage are each equipped with range, water boiler, sink and bath.

There were installed nine 1,000-gallon kerosene fuel tanks and one 500-gallon gasolene fuel tank on a concrete foundation and with connecting pipes and leads to engine fuel tanks.

The dam was made permanent and a pipe line was led to the power house 396 feet distant. A 1½-H.P. gasolene-driven pump and four 500-gallon engine circulating-water tanks were installed in the power house. One 500-gallon tank was installed in the attic of the dwelling and another in the attic of the cottage. A 5,000-gallon redwood water tank was set up near the power house. Necessary pipe leads were made between the pump, tanks, engines and water systems of the dwelling and cottage. The pipe was laid under ground and provision made for draining to protect against freezing. A lead was also made for flooding the ground plate.

A cesspool was dug with sewer connections to both dwelling and cottage.

The expedition also built a coal house, an oil house and a small house for the 5-H.P. gasolene engine used with the railroad. The installation of the railroad was made permanent.

The operating room contains one 10-kw., 500-cycle, quenched-spark, Telefunken radio set, the 3-kw., rotary-spark, auxiliary radio set and a small storage-battery sending set.

The power house contains two 25-H.P. Fairbanks-Morse kerosene engines driving, by leather belt, two 10-kw., 500-cycle alternators with exciters. These were all mounted on one concrete foundation weighing 114 tons. A 10-H.P. gas engine, after being used to drive a saw bench, was installed on a 6½-ton concrete foundation, and, by leather belt, drives the 3-kw. auxiliary set alternator, with its exciter mounted

on same shaft. The exciter is also used for electric lighting and charging storage batteries.

The leads from the power house to the dwelling were run through the tunnel under the passageway between the two buildings and are easily accessible.

The power house also contains a lathe which can be driven by the pump or the 10-H.P. engine, and a large carpenter's work bench.

The ground plate consists of sheet copper covering a surface of 420 square feet and buried at an average depth of 4 feet, in moist clay and 20 bushels of charcoal. It is connected to all ground leads and protection devices in the dwelling and the power house. The ground was tested when operating with the 10-kw. set and no potential was obtained at the base of the aerial.

The main aerial is of the flat-top type, 12 wire, length 550 feet, spread 40 feet, down leads at the eastern end, bunched near the top, and stopped together all the way down. There was also installed a vertical aerial for the auxiliary set, 4 wire, length 250 feet, leading to the operating room from a point on the eastern mast 50 feet below the end of the main aerial. There is a single-wire receiving aerial 6 feet above the main aerial with a single down lead at eastern end. Switches were installed to allow the use of any one or all aerials, or any combination of them. The main aerial was led into the dwelling through electrose tubes and further insulated by micanite plates in roof of the attic and ceiling of the operating room.

The station was furnished with complete equipment, including necessary furniture, table and kitchen ware, linen, stoves, a library of 400 books, technical and fiction; a phonograph with records, rifles and shot gun, and a complete outfit of machinist's, carpenter's and electrician's tools. A 44-foot motor boat, 35 H.P., was also furnished the station, but this was totally lost in a storm on September 12, six days after the expedition left. There was also furnished a sea-otter boat

and a 16-foot skiff. The station was given sufficient supplies to last two years.

Unalga was commissioned at 12:01 A. M., September 1st, and communication was established with Mare Island, Eureka, North Head and Sitka. The following day daylight communication was established with Dutch Harbor and Pribilof, with both main and auxiliary sets. The station clears with the Pacific coast stations every night, which includes forwarding the daily weather report of the observer at Dutch Harbor.

On September 3d the working party was sent back to the ship with bedding and provisions, and 38 men were brought ashore the morning of the 4th to gather up the small amount of material remaining. A violent storm began without warning at 10 A. M., and it was impossible for the men to return to the ship, which hurriedly put to sea. The men were marooned ashore until noon of the 6th, when the *Nero* returned. Provisions belonging to the station force were used and replaced later. The masts, buildings and aerials underwent a severe test without injury. The expedition left Unalga at 4 P. M., September 6th, for Dutch Harbor. Six days later, the 12th, the motor boat was lost and the wharf carried away in a heavy storm.

The cost of the construction work at Unalga was $55,590.

DUTCH HARBOR.

On July 15th the *Nero* left Unalga with 23 of the working party and arrived at Dutch Harbor the same date. Most of the material for Dutch Harbor had been landed there on other days when the *Nero* was driven to that port to seek shelter from storms, and natives had been employed to transfer material from the landing to the station.

The working party remained at Dutch Harbor 14 days, during which time two 150-foot wooden lattice masts were erected to replace the ones blown down by hurricane in October, 1911. The roof and sides of the building were shingled, storm windows installed and the interior finished with

plaster board and battens. This rendered necessary the removal and reinstallation of the radio apparatus. Three 1,000-gallon kerosene fuel tanks and a 500-gallon gasolene fuel tank were installed. The piping for the kerosene tanks was so arranged that the tank could be filled from drums by air pressure working through a relief valve set at 20 pounds.

Dutch Harbor station has a 5-kw., 500-cycle, quenched-spark Telefunken radio set. Duplicate 12-H.P. Meitz and Weiss kerosene engines are used to drive the alternators. The radio set, engines and generators were thoroughly overhauled. There was installed a 6-wire, flat-top aerial, length 400 feet, spread 30 feet, with four down leads at the north end, bunched near top and stopped together all the way down. A single-wire receiving aerial was installed 5 feet below the main aerial, with single down lead at north end. This aerial is also used for sending on short wave lengths. Switches were arranged for using either aerial, or both at the same time. The station was furnished with necessary additional equipment and sufficient supplies for two years.

The station was recommissioned the night of July 29, 1912, and communication was immediately established with Sitka and Cordova. In August intermittent communication was established with Mare Island and Eureka, Calfornia.

Between 2 and 4 A. M., September 5th, a southwest gale with a reported velocity of 100 miles per hour during puffs, blew down the south mast. The storm also did considerable damage in the village of Unalaska and some damage at Dutch Harbor. On the afternoon of the 5th the station force had rigged a temporary aerial to the remaining mast, and were again in commission.

The *Nero* arrived at Dutch Harbor on the night of the 6th, and on the 7th the wreckage was cleared away and an 80-foot ship's mast set up and new aerials installed similar to the ones which had been carried away. The telephone line connecting the station with Dutch Harbor and Unalaska had suffered from the storm and this was also repaired.

The cost of new construction and the repair work at Dutch Harbor amounted to $3,500.

ST. GEORGE.

The *Nero*, with a working party of 19 men, arrived at St. George, Pribilof Islands, the night of August 3d and left the night of the 6th. No work was done on the 5th as it was impossible for the working party to land. The installation at St. George is only for communicating with the larger station at St. Paul and with ships in the immediate vicinity.

The set consists of a 1½-H.P. gasolene engine, driving a 20-volt, 25-ampère, D. C. generator, which can be used for sending, or through a rheostat for charging storage batteries. Two Edison type B-4 batteries were furnished, also for sending, with a 4-inch and a 2-inch coil, either of which can be used. A small spark gap and a sending key completed the transmitting apparatus. For receiving, there was a Perikon detector, Murdock tuner, a Marconi and Shumaker condenser and standard phones. A 60-foot galvanized iron-pipe mast was erected and a single-wire antennae, 270 feet long, led from the top of the mast to the operating desk. The ground plate was buried about 5 feet, near the operating desk.

St. George is manned by one naval operator during the summer months and at other times the Government school teacher acts in that capacity.

The cost of work at St. George was $700.

ST. PAUL, PRIBILOF.

The *Nero* and working party were at St. Paul, Pribilof, from August 7th to August 26th. Sixteen concrete anchors of 9 tons each, were installed for the two 225-foot lattice masts of the station, and new rigging was set up for each mast. The main aerial is a flat-top 6 wire, length 400 feet, spread 30 feet, 4 down leads at eastern end, bunched near top and stopped all the way down. There is also a single-wire

aerial for receiving and for sending on short wave lengths, 5 feet above main aerial.

Kerosene and gasolene tanks were installed, similar to those at Dutch Harbor. The old 60-cycle radio set was removed and a 5-kw., 500-cycle, quenched-spark Telefunken radio set was installed. Two 12-H.P. Meitz and Weiss kerosene engines were installed on a concrete foundation. Storm windows were fitted on both dwelling and power house and arrangements made for natives to shingle sides and roofs of the buildings. This work was completed after the departure of the expedition. The interiors of the buildings were finished with plaster board and battens and a small bed room was changed into a bath room. A cesspool was dug and a sewer was put in. Additional equipment and supplies for two years were furnished the station.

The nearest location for obtaining fresh water was 2,700 feet from the station. The expedition agreed to install a 5-H.P. gasolene-driven pump and to furnish necessary material and lay a pipe line from the well to the station, provided the island natives dug the well and the trench for the pipe line. This agreement was reached in view of the mutual benefits to both the village and radio station. The natives failed to perform their part of the work in time for the expedition to do its share before the date of departure. When the *Nero* sailed two men were left to assist in the completion of the work; these men joining the expedition later at Dutch Harbor via the United States revenue cutter *Tahoma*. The work was finally completed and the pipe line extended to the village, a quarter of a mile beyond the station.

The station was recommissioned August 24th, and the first messages with the new set were exchanged with Mare Island, a distance of 2,187 miles. The same night communication was established with Eureka, California, Sitka and Cordova, and the next day daylight communication was effected with Dutch Harbor and the temporary set at Unalga. After the departure of the expedition the St. Paul and Dutch Harbor stations exchanged signals during daylight, using only

storage-battery sets with small spark coils. This distance was 250 miles.

The location of the St. Paul station is excellent for radio communication and better results are obtained than at any other Pacific station. The absence of any high land, the good ground connections, the sweep over water before any land intervenes, are potent factors in making its work so effective.

The cost of the new installation and repairs at St. Paul was $10,395.

CORDOVA.

The station at Cordova is located at Cape Whitshed, 9 miles from Cordova, where the *Nero* anchored on September 14th. A large barge was rented and material was loaded on it and sent to the station. The working party had to wait for a high tide before the barge could be beached close in to the station. In the meantime, the required material was dragged about 100 yards over the mud flats.

A 10-kw., 500-cycle, quenched-spark Telefunken radio set was installed after removing the old 60-cycle set. Duplicate 25-H.P. kerosene engines, similar to those installed at Unalga, were set up on a 92-ton concrete foundation. An extension was built to the power house for holding the fuel and circulating water tanks for the engines. Three 1,000-gallon kerosene fuel tanks were set up with necessary piping.

Between the 200-foot steel towers was installed a flat-top aerial, 8-wire, length 500 feet, spread 40 feet, with 6 down leads at the south end, bunched at top and stopped together all the way down. There was also installed a single-wire aerial, "T" type (15 feet above main aerial), for receiving and for sending on short wave lengths. The aerial at Cordova is liable to be carried away during winter by ice and snow, and facilities were furnished in order that repairs could be made quickly.

A three-room cottage, with shingled roof and sides, was built for the electrician in charge. The inside was finished

in the same manner as the cottage at Unalga. The station had a 10-H.P. motor boat, which was overhauled and an extension built to its cabin. In view of the probable damage to this boat from floating ice during the winter, a boat house was built, with ways, and a cradle and windlass were furnished for hauling the boat out of the water into the boat house.

The water system of the station was completely renewed and a number of small repair jobs and minor alterations were completed. In addition to the new radio set, other necessary equipment and some supplies were left at the station.

The station was recommissioned October 7th, 1912, and communication immediately established with Eureka, California, Unalga, Dutch Harbor and Pribilof. Daylight communication was established with Sitka, and later on with Kodiak, and at a later date night communication was established with Mare Island and North Head. The *Nero* left for Kodiak on October 10th, twenty-six days after arrival at Cordova.

The cost of the new installation and the repair work at Cordova amounted to $20,200.

KODIAK.

The Kodiak station is located on Woody Island, about two miles from the village of St. Paul on the main island. With the exception of the masts, the station had been totally destroyed on June 8, 1912, when struck by lightning during eruption of Mt. Katmai. There was a depth of ashes, or rather powdered pumice, for about 8 to 12 inches all over the land. The only article left in the ruins of the old station which was of any value was a 15-H.P. gas engine, which was returned to Mare Island by the *Nero* and afterwards repaired.

All material for the new station was landed in three and a half days. A combined dwelling and power house was erected, with shingle roof and sides, storm windows fitted throughout and a glass porch built outside the operating room. The

inside of the building was finished with plaster board and battens. The dwelling was fitted with electric lights and equipment for all rooms, including kitchen and bath. A water system was installed with a 500-gallon tank in the attic, a 2,500-gallon redwood tank just outside the power house, and a 2-H.P. gasolene-driven pump. Water was pumped through pipe line from a well near the station.

Two 12-H.P. Meitz and Weiss kerosene engines were installed on a 44-ton concrete foundation, with 5-kw., 500-cycle alternator. A 3-kw. generator was set up for electric lighting and charging storage batteries. The engines were equipped with constant-burning lamps and necessary tanks. Three 1,000-gallon kerosene fuel tanks and one 500-gallon gasolene fuel tank were set up in the same manner as the ones at Dutch Harbor.

In the operating room was installed the 5-kw., 500-cycle, quenched-spark, experimental radio set, designed by Expert Electrical Aide George E. Hanscom. The set permitted rapid changes in wave-length settings from 300 to 6,000 meters, the latter being greater than obtained in any other station on the Pacific coast. A new ground plate was put in and connected to the ground of the old station, and, as usual, to all ground leads and protection devices.

New rigging was set up on both of the 225-foot wooden lattice masts and a main flat-top 8-wire aerial was installed, length 375 feet, spread 40 feet, 4 down leads at eastern end, bunched at top and stopped together all the way down. There was installed a single-wire aerial, "T" type (ten feet above main aerial) for receiving and for sending on short wave lengths. The station was furnished with a lathe, equipment for the dwelling and power house, and supplies for about one year.

An auxiliary station was established in the town of St. Paul to facilitate local communication. This station has a 60-foot mast, with a 4-wire vertical aerial, length 75 feet, spread 6 feet. A radio set, which was the usual storage-bat-

tery outfit, was installed in a one-room cottage built on the U. S. Customs Reservation.

The new Kodiak station was commissioned at 12:01 A. M., November 1st, eighteen days after the arrival of the expedition. Immediate communication was established with Cordova, Sitka and Unalga, and the next night with other Alaska stations and with Pacific coast stations as far south as Mare Island. On November 2d, the Mare Island operator, without previous intention or notification, tested all the new Alaska installations by calling in succession, Cordova, Kodiak, Unalga, Dutch Harbor and Pribilof, and obtained the daily weather report from each. The time required was less than thirty minutes.

The *Nero* left Kodiak November 5th, 1912, for Sitka.

The cost of the Kodiak station, including an estimate of $5,000 for the purchase of the land on which the station was built, was $23,283.

SITKA.

The expedition was at Sitka from November 8 to 13, 1912. A rotary-spark gap was installed, the condenser capacity increased, and minor repairs made to the apparatus. An 8-wire flat-top aerial was installed, length 400 feet, spread 40 feet, with four down leads at one end. There was also installed a single-wire aerial, "T" type, 200 feet vertical, 100 feet horizontal, for receiving and for sending on short wave lengths.

The plant at Sitka consists of duplicate 35-H.P. gasolene engines and 20-kw. separately excited alternators. The radio installation is a 10-kw. rotary-spark gap set, rated for the output used.

The cost of the work at Sitka was $540.

The *Nero* returned to Mare Island on November 23d, 1912, after a cruise of six months and three days. The work originally planned for the expedition had been carried out, although at one time it was feared that a postponement of some of the work for another year would be necessary on account of the bad weather encountered and the rapid approach of winter.

The total cost of the expedition, including the subsistence of the personnel, was $136,450. This amount, however, did not include the maintenance cost of the *Nero* nor the pay of her officers and crew.

During the cruise the health of the working party in general was good, in spite of exposure to inclement weather and the difficulty of obtaining fresh food. There were no serious accidents to the personnel. The material suffered from storms, but only one-half of one per cent. of the material carried was lost.

At Unalga, Dutch Harbor and Pribilof, the revenue cutters of the Behring Sea Patrol Fleet assisted materially in local transportation of men, mail and some supplies. An effort was made by the *Bear* and *Tahoma* to raise the sunken Unalga motor boat after the storm of September 12th, but a succession of heavy gales prevented any possibility of success. The Government agents at St. Paul and St. George also rendered valuable assistance to the expedition when at those islands.

The importance of the Alaskan stations will increase from year to year, particularly in view of the law authorizing them to handle commercial business. Districts which had no communication with the United States other than intermittent and irregular steamers are now in telegraphic communication, via radio to Eureka for points in California and to North Head for points outside of that State. Unalga can be a relay station from the United States to Japan and Siberia. Communication with the former has already been effected via Otchisi. Dutch Harbor is the farthest point to the westward at which a weather observer is stationed, and the daily reports from that station are of great assistance to forecasters in making weather predictions for the Pacific coast.

The most important work of the Alaskan stations is communication with vessels crossing the Pacific by the northern route, and with those sailing between Seattle and Nome and southeastern Alaska ports. Times of arrival and departure, private and business messages from passengers, the state of the

weather for transmission to other vessels, comprise the major portions of this work.

The dangers to navigation in Alaska are many, and these are increased by heavy fogs during the summer months. Should a vessel be wrecked or disabled while in Alaskan waters, her commander can communicate with the nearest radio station and be assured that assistance will be received from the nearest vessel or port as soon as possible. The saving of one vessel will justify the cost of the installation of the Alaskan stations. The rescue of all or some of the lives on board, even if the vessel is lost, cannot be reduced to terms embracing cash values.

Naval operators in Alaska must and do realize the necessity for constant attention when on watch, and, in spite of the lonely life and other hardships, it is believed the efficiency of the personnel will always be in advance of the efficiency of the installation, even though the latter is excellent.

NOTES.

A THEORY OF SURFACE CONDENSATION.

Discussion of Mr. D. B. Morison's article in the February, 1913, Journal by William Weir.

In the first place, Mr. Morison lays great stress on the importance and value on Prof. Weighton's experiments and claims that they definitely established the correctness of certain principles. While having no desire to depreciate the cost and elaboration of Prof. Weighton's research, I totally disagree with Mr. Morison as to its value and especially as to its effect on present-day design. The paper describing these experiments dealt with the comparative performances of two condensers, one an old-fashioned rectangular condenser of small size, unrepresentative of practice at that date, and the second a condenser with certain special features of design. The trials showed that the new condenser gave a better performance than the old one. Prof. Weighton and Mr. Morison attributed the improvement to:

1. "Compartmental drainage of the feed water, so that the surfaces in the lower compartments are not impeded in their condensing action, the importance of this feature being apparent, and its influence being seen on the results."

2. "Abolition of the steam space, and the substitution of a passage of such shape and section as will secure the distribution of the steam over the whole length of the tubes."

3. "Tubeless passages connecting the compartments."

Prof. Weighton stated in this paper, in at least four different instances, that the augmented efficiency of the surface was due to the early interception and removal of the feed water. He also said that the higher hot-well temperature and the smaller amount of condensing water and surface were both due to the same cause—the compartmental drainage of the condenser. In the discussion on this paper Mr. Morison stated that the novel features of the apparatus consisted of sectional drainage, maintenance of parallel steam lines of flow, and alteration of direction of flow in tubeless passages. He also stated that there was no novel feature on the water side, as the water flowed through the nests of tubes in series at speeds adopted in modern practice.

The correct value of Prof. Weighton's paper can be measured by the extent to which the alleged improvements have become embodied in modern designs. In the last four years I have personally been associated with the design of condensers for about nine million horsepower, embracing the most important examples in marine practice, and in not a single case were the condensers divided into compartments, nor were they sectionally drained; they had all more or less steam space, and none of them had tubeless passages.

The factor which was mainly responsible for the better performance of the second condenser was undoubtedly the increase in velocity of flow of the circulating water through the tubes.

With reference to Mr. Morison's remark regarding the provision for dealing with large quantities of air leakage, resulting from accidental leakage into the system, I simply say that I have had no experience of

accidental leakage which a normal air pump could not handle satisfactorily enough under the circumstances until the accident was repaired, and I think it quite unnecessary to provide air pumps to maintain the highest degree of vacuum under other accidental conditions.

The most important part of Mr. Morison's paper was where he stated that "in practice the crucial problem in the attainment of high vacuum is how best to treat the small quantity of air which normally enters the condenser with the steam," and he submitted a theory whereby a solution might be found along the simple lines of elementary first principles. Now, in the first place, careful observation of phenomena is generally more helpful than theory in offering solutions of problems. The best way to treat the small quantity of air in a condenser, to attain high vacuum, is to fit an air pump to the condenser capable of removing the air, and I am afraid Mr. Morison would find difficulty in obtaining any other solution of the problem. Dealing now with the theory and its basis, Mr. Morison said Mr. J. A. Smith carried out experiments to determine the effect of air on heat transmission through a condenser tube, and certain results were quoted from these experiments. I can not emphasize too strongly that any theory or practice based on or affected by these experiments would be absolutely valueless, while, generally speaking, the results had no bearing whatever on surface-condenser design or practice. Mr. Morison himself confirmed the unfortunate effect of these fallacious results, when he stated that a film of air of minute thickness had a degrading effect on the heat transmission of a condenser tube. In the first place, there were no films of air nor were there any films of steam in a condenser. There was a mixture of air and steam or air and aqueous vapor, each occupying the whole space. It would, no doubt, be remembered that one of his essentials of design was that the steam path must be such as to maintain the velocity of steam flow, and this was a most important factor. The relative weights of steam and air played no appreciable part in these condenser phenomena. In Mr. Smith's experiments the steam conditions were practically static, and accordingly had no direct connection with condenser phenomena, and the basis flaw of Mr. Morison's theory was that he always considered a condenser in a static condition rather than in a working condition. The effect of air in a working surface condenser was not in any way due to a resistance imposed on heat transmission by its presence and through being a poor heat conductor, but solely and simply to the physical conditions of temperature and pressure, caused by its admixture with aqueous vapor. I totally disagree with Mr. Morison's illustrative hypothetical diagrams showing air distribution, and only fear that these might lead to a further misconception of the real conditions. In a working condenser the phenomenon consists of a mass of cooling surface working at high transmission rates, with a more or less clearly defined air zone wherein the total transmission per square foot, not per degree difference per square foot, was impaired, due to the physical conditions of temperature and pressure. Accordingly, if the essential condition of uniform steam flow be maintained in a condenser design the contour of the condenser became a wedge, and having obtained that all else was dependent on the air pump, assuming fixed circulating-water conditions. The function of the air pump was to reduce the air zone level and to allow the working part of the condenser to extend itself, and only the air pump could perform this function. Mr. Morison's remarks regarding eddies and means of preventing air particles making multiple appearances in any one plane could hardly be accepted as serious when the actual velocities were remembered. The idea of an air particle lingering in a modern condenser with uniform steam flow is an effort of imagination of which I am incapable. The worst eddies in a condenser are those caused by the tubes. Any other eddy troubles are caused by the presence of baffle plates or deflectors.

Referring to Diagram I, showing reduction of power by fall in vacuum.

Almost everything in engineering is a compromise, and while on one hand a fair percentage of improvement in steam might be realized, on the other hand one has to consider the increased circulating-pump consumption, the increased weight and space for air and circulating pump, the increased weight and space of main condensers, the effect of reduced hotwell temperature, the increased weight and space of the low-pressure turbines to utilize satisfactorily the higher vacuum, so that in the end, even for turbines, a practical limit in reduction of back pressure might perhaps be regarded as desirable.

THE CORROSION OF DISTILLING CONDENSER TUBES.

By Arnold Philip, B. Sc., A. M. I. E. E., Assoc. R. S. M. (Admiralty Chemist).

Distilling apparatus is largely used both on merchant steamers and on ships of the Royal Navy for the purpose of preparing fresh water, not only for washing and cooking purposes, but also for supplying the make-up feed water for the steam boilers.

The distilling apparatus consists in general of two parts. Firstly, a cylindrical boiler or evaporator which is fed with sea water either by hand or automatically. In this sea water a steam-heating coil is either partially or totally immersed. Through this steam-heating coil a current of high-pressure boiler steam (primary steam) is passed at a temperature of from 240 to nearly 400 degrees F.

The raising of the sea water surrounding the coils to this temperature causes it to boil, and the steam (known as secondary steam) which is thus evolved from it passes over into the second part of the distilling plant. This is known as the distilling condenser, and is in general very similar in construction to the forms of main and auxiliary condensers used on steamships.

There are, however, one or two differences between the construction of the distilling condenser and the auxiliary and main condensers as now generally used in H.M. service which should be noted. In the distilling condenser the steam passes through the condenser tubes and the cooling water circulates outside them. This is the reverse of the usual practice in main and auxiliary condensers. Secondly, the tubes in the distilling condenser are usually vertical and expanded into the tube plates, whilst in the main and auxiliary condensers they are horizontal and pass through watertight glands in the tube plates.*

It is interesting to remark here that, whether on account of the fact that the distilling condenser tubes are vertical, or whether because they are expanded at their ends into metallic contact with the tube plates, or whether because the sea water circulates outside the tubes inside a steel casing instead of inside the tubes, no case is known to the author of corrosion trouble occurring to the tubes in these condensers on their sea-water side. This statement that corrosion is absent on the sea-water side of condenser tubes in a distilling condenser merely represents the author's personal experience and the results of the inquiries he has been able to make, and to this extent it appears to confirm the views already expressed by him in previous communications to the effect that by far the greater part of the trouble with the corrosion of condenser

* It should be stated that distilling condensers of other patterns in which the cooling water passes through the tubes and also in which the tubes are horizontal are used, but the type most generally employed until recently is that here described, and the remarks as to corrosion on the sea-water side of distilling condenser tubes must be understood to apply to this form of condenser only.

tubes with sea water is due to the presence of carbon or other electrically conducting relatively electro-negative body deposited along the inside of the tubes at the bottom, and that on the other hand practically all the other less important corrosive actions can be stopped by the use of masses of properly connected iron and steel or perhaps aluminum and zinc as protective metals.

Notwithstanding this freedom from corrosion of the tubes of distilling condensers upon their outer or sea-water surfaces, very great difficulties have been experienced from the corrosion of these tubes upon their inner or steam surfaces.

From time to time in the past, what in the aggregate is a large number of cases have occurred in which very serious corrosion has taken place on the steam side of the condenser tubes of sea-water distilling apparatus.

The trouble usually first made its appearance by the presence of what was believed to be priming in the distilling evaporator. This was indicated by the fact that the water from the distilling condenser gave a precipitate or cloudiness when tested with silver nitrate solution. More marked symptoms of corrosive action were indicated by the distilled water possessing a disagreeably metallic taste, and reports from various ships showed that soap used with such water sometimes gave a light bluish-green curd. In one case the distilled water, after boiling in the ship's "copper," was used for making tea, and gave a black-colored infusion, whilst the ship's "copper" (actually made of iron) became coated internally with metallic copper. In short, the distilled water contained copper, and on examining the distilling condenser tubes they were found to be markedly corroded on their steam side. The corrosion was so active that the distilled water in once passing through the tubes dissolved sufficient copper from them to produce the troubles referred to. The cause of the presence of the copper was at first, as stated above, attributed to the priming of the sea water in the evaporators, and the action of the sea salt thus passing over into the distilling condenser tubes was looked upon as causing the solution of the copper. This view appeared to be confirmed by the fact that the usual silver-nitrate test of the distilled water seemed always to demonstrate the presence of salt.

Early endeavors to reduce the effects of the supposed priming were made by coating the insides of all the condenser tubes and other steam pipes with tin, whilst steel and zinc vapor baffle plates and protector blocks were placed in the head of the evaporator and in the vapor pipe connecting the head of the evaporator with the distilling condenser. Both these remedies were found to effect a temporary cure, but unfortunately after a while the same troubles reappeared. The degree to which these difficulties, due to the corrosion of the tubes and the presence of copper in the distilled water, were noticed was found to vary a good deal in different ships, and even in the same ship marked fluctuations occurred from time to time.

The author's first examination of a case in which corrosion of distilling condenser tubes had been observed was in August, 1904, and on looking through previous correspondence upon this and other similar corrosions he found, in a letter written by an engineer officer in 1903, the suggestion that as part of the heating surface of the primary steam coils in a particular evaporator was under normal conditions of working left above the sea-water level, it was possible that this getting quickly coated with scale and consequently overheated might evolve hydrochloric acid from the magnesium salts found in the deposit, and that it was the acid thus evolved which caused the corrosion observed in the distilling condensers.

This observation appeared to be a very probable explanation of the corrosive troubles which had been noticed, and steps were at once taken to fully examine the particular case then under question from this point of view.

It was found that generally evaporators may be divided into two classes—namely, those containing drowned heating coils—(that is, in which the primary steam coils are completely immersed in the sea water or brine which they are used for heating), and those in which the primary steam coils are only partially covered with water.

Experiments made on board a ship which was fitted with evaporators both of the drowned-coil type and the partially-exposed coil type showed that when connected to the distilling condenser the drowned-coil type evaporator always made water practically quite free from copper, whilst the evaporator with the exposed heating coils when connected to the same condenser gave water which contained copper.

The brine from both types of evaporator was then examined, and it was found that from the drowned-coil type it had a very faintly alkaline reaction to phenolphthalein, whilst the brine from the evaporator having the partially exposed heating coil possessed a very strongly alkaline reaction to the same indicator.

On opening up an evaporator and examining the inside surfaces of the walls of its casing, it is usually found that the lower portion, which contains the sea water or brine which is undergoing evaporation, is coated all over with a brilliant white incrustation, and this white incrustation extends upwards above the level of the brine, as far in fact as the water can splash when boiling. Above this region, however, the inside surfaces of the head of the evaporator (if it is not of the drowned-coil variety) are usually much corroded and colored a deep orange-brown. The tops of the steam coil, as far as they project above the surface of the brine, are also thickly coated with white incrustation. Chemical analysis shows this incrustation to be a mixture of calcium sulphate and basic sulphate and chlorides of magnesia, together with a considerable proportion (2.35 per cent.) of magnesium hydrate. This incrustation is normally heated up to practically the same temperature as that of the primary steam—that is, to a temperature considerably over that of the boiling brine, and it is at the same time constantly splashed with the boiling brine itself. Under these conditions the magnesium chloride present is decomposed, hydrochloric acid is evolved with the steam, and a strongly alkaline incrustration and also an alkaline brine are formed.

The chemical tests on the white incrustation and the alkalinity of the brine itself clearly show that some acid must have been driven off by the process of treating the sea water in the evaporator. Sea water is itself quite neutral to phenolphthalein to commence with, and the amount of alkaline base found to be separated was much greater than could be explained by any other hypothesis than that hydrochloric acid had been driven off.

In order, however, to demonstrate beyond dispute that hydrochloric acid was actually evolved, it was considered desirable to prove its presence in the distilled water.

This was a matter of some little difficulty, on account of the very small amount in which the acid was present, namely, from 0.1 to 0.05 or less parts of HCl per 100,000. To determine the acidity of such a weak solution by direct titration is not a satisfactory procedure. Hydrochloric acid is so volatile that it is not possible to concentrate it by evaporation. The distilled water, besides containing hydrochloric acid, also contains sodium chloride due to slight priming, and also chlorides of copper, tin, and zinc, formed by the acid water flowing through the condenser tubes. The quantities of these metallic chlorides which are present in the water are of about the same order of magnitude as the amount of hydrochloric acid itself. As these salts lose HCl on evaporating to dryness, the method of determining the difference between the total chlorine present in the water and then the chlorine present in the solid residue after distillation to dryness does not give the correct amount of free hydrochloric acid present.

The method of test which was finally adopted was to add sufficient of a saturated solution of pure sulphate of silver (7.72 grams per litre) to the water. This causes a precipitate of silver chloride to be formed, and free sulphuric acid equivalent to the amount of free hydrochloric acid present is liberated; after heating to coagulate any silver chloride formed, it is filtered off and a large bulk of the water thus treated is evaporated down to a small bulk. This then contains all the free sulphuric acid. To demonstrate that it was free sulphuric acid a portion of it was dropped on a piece of lump sugar, and this was heated for an hour in a water oven at 210 degrees F. The presence of the sulphuric acid was demonstrated by the blackening of the sugar as the acid became concentrated, whilst a similar piece of sugar treated as a control with a residue from ordinary distilled water to which the same amount of the silver sulphate solution had been added gave no blackening. In order to be certain that the silver sulphate was itself quite free from acid before use it was washed by boiling up with successive small quantities of distilled water, and the washed salt was employed for making up the silver-sulphate solution used in the experiments. The actual determination of the amount of the free acid present was carried out by a titration of the concentrated residue, obtained from a further portion of the water under examination, after it had been treated with the pure silver-sulphate solution as described above.

Tests carried out in this manner showed that the water obtained from distilling plant which caused corrosion of the condenser tubes, and which contained copper, invariably also contained free hydrochloric acid.

The amount of copper present in such waters is conveniently estimated by a calorimetric test, using potassium ferrocyanide in comparison with standard solutions of a copper salt.

The results of some actual tests made upon samples of water obtained from two distilling plants, in which one had an evaporator with the upper portion of its primary steam coils above the surface of the boiling brine, whilst the other had its steam coils drowned (that is to say, completely immersed beneath the surface of the brine), are given in the table below.

Type of Elevator	Coils exposed above surface of brine.		Drowned coils.	
Rate of working evaporator, in tons of distilled water per diem	16	8	13	5
Pressure of primary steam, in pounds per square inch absolute	215	60	40	25
Corresponding temperature of primary steam, degrees F	337.4	292.5	267.1	240
Pressure of secondary steam, in pounds per square inch absolute	30	27	18	15.5
Corresponding temperature of secondary steam, degrees F	258	244.1	221.9	214.5
Copper, in grains per gallon, present in the distilled water obtained	0.32	0.56	0.005	0.008
Reaction of the brine in evaporating to phenolphthalein	Strongly alkaline.		Very faintly alkaline.	
Reaction of distilled water to methyl orange	Faintly acid.		Faintly acid.	

At the date at which these results were obtained, 1904, the method of determining quantitatively the amount of free hydrochloric acid present had not been worked out, but tests for free hydrochloric acid made upon other samples of distilled water by the silver-sulphate method a few

months later showed, as had been stated above, that this acid in the free state is always present in the distilled water obtained from an evaporator in which the primary steam coils are exposed in the secondary steam space above the brine.

From the results given above, however, it is apparent that the rate at which an evaporator is worked or, in other words, the temperature of the primary steam, caused the amount of copper in the distilled water, and therefore of the hydrochloric acid formed in the evaporator, to vary. The more the evaporator is pressed, the greater the amount of hydrochloric acid formed in a given time. There is further some evidence to show that the evolution of hydrochloric acid from the salts in sea water may not only be caused by the heating of the saline incrustation on the heated steam coils, but that this acid may also, but to a much smaller extent, be given off from the incrustations formed on the lower sides of the evaporator shell, and also possibly even from the brine itself, and in a given form of evaporator the formation of free hydrochloric acid is favored by the high temperature of the primary steam coils and the strong concentration of the brine. To obtain the least quantity of hydrochloric acid from a given evaporator it is therefore necessary to avoid too great a concentration of the brine by suitable adjustment of the sea-water feed and brine cocks, and also, and most effectively, to reduce the rate of evaporation as much as possible. Both of these methods of remedying the trouble are, however, faulty, for they both tend to render the evaporator inefficient. The best remedy is to only use evaporators with drowned steam coils.

The passage of copper into the boilers with the feed water is, of course, most undesirable from the point of view of boiler corrosion, and it has been proposed to avoid this by passing the distilled water through a scrubber of large granulated zinc or small zinc blocks. The feed tank into which the water from the distilling condensers is pumped is usually fitted with zinc protectors attached to its walls. In the absence of zinc scrubbers, if owing to defects in the evaporators the presence of copper in feed water is unavoidable, its harmful effects may be best minimized by keeping the water alkaline with lime, for this causes the objectionable soluble copper salt to be converted into the comparatively harmless insoluble precipitate of cupric hydrate, but this palliative with boilers running at high pressures, and in the presence of organic oils, must be regarded with distrust.

From what has been stated above it is evident that the measure taken to prevent the presence of copper in the distilled water, by tinning the condenser tubes, causes the temporary disappearance of the copper, because tin, instead of copper, is dissolved from the inner surfaces of the condenser tubes, and this metal is not readily detected in the distilled water, whilst directly the tin has been removed by the acid water the copper will once more make its appearance. The use of steel or zinc baffle plates in the head of the evaporator, and zinc protector blocks in the vapor pipe, also cause a temporary removal or diminution of the amount of copper in the distilled water, due to the fact that the baffles, etc., remove, or partly remove, the hydrochloric acid from the evaporator steam, but as the plates become rapidly corroded, they become inefficient and the trouble again appears.

The serious nature of the corrosion which may occur, due to the acid evaporator steam, is well shown from the following extract from a description by an engineer officer of what occurred in a ship using evaporators with steam coils which were not drowned:

"The brass valves and valve seatings wear in the most excessive manner, in fact it appears as though one or more constituents of the metal were eaten out of it altogether, and only a hard mass left; this remark applies particularly to the vapor valves, through which the secondary

(gained) steam passes. * * * It is no new thing, but has been going on for the past three years. The fresh water made always has a metallic taste, and when worked with soap turns a very slightly pale greenish color; but the fresh water made by a second evaporator which has drowned coils is perfectly fresh, tasteless and good, and gives no color with soapy water, although the steam is condensed in the *same* distilling condenser as the unsatisfactory water from the other evaporator."—"Page's Weekly."

Read before the Institute of Metals.

RESISTANCE AND SPEED OF SUBMARINES.

It is well known to those conversant with the stream-line theory, that a body wholly submerged in a frictionless fluid, and creating no waves, can be maintained at a uniform speed of motion, when once that motion has been established, with no further expenditure of power; that is to say, the stream-line motion when once set up, will be indefinitely maintained until some force be impressed upon the submerged body to change the uniformity of its motion. This seems rather an extraordinary theory, but it is obviously true when we consider that the frictionless fluid is displaced by the advancing forward portion of the submerged body, only to be replaced by flowing in around the after part, resuming its original state of rest after the passage of the body, and therefore demanding no force to maintain an impressed motion, since none is imparted to it. This theory is a purely hypothetical one, for there is no such thing as a perfectly frictionless fluid, nor an infinitely deep and wide ocean, in which the body may be submerged so that there is no wave making. There can, therefore, be no motion such as is postulated without both surface friction and wave-making disturbance, and also to a greater or less degree, losses from eddy-making; all of these acting together remove the question of the actual resistance of submarines far from the ideal conditions laid down.

It is necessary in considering the comparison between the resistance of submarines when floating on the surface and when partly or wholly submerged to be able to form some estimate of the relative values of each of these factors in the two conditions. At the first blush, it seems from the above that the submerged condition will exhibit a lower resistance than the floating condition for the same speed, since wave making when on the surface forms a large proportion of the total resistance, and when submerged must bear a very much smaller ratio to the whole resistance, for we may count out eddy making as of relatively small importance in both cases. This is quite true so far as wave making alone is concerned, but when the relative forms and amount of immersed surface are considered, the comparison takes on a totally different color, for while actual surface disturbance or wave making is much less in the submerged condition, the skin friction is very much increased. It must be premised that the submarine we are considering is one of the modern type with a forward body differing very little indeed in shape from that of an ordinary torpedo boat or destroyer; that is to say, having a decked forecastle, which is almost a necessity and certainly a great advantage, for surface navigation. This deck is carried aft to a greater or less length—depending on the type—and has a breadth slightly less than that of the main body of the vessel. On this deck is an erection which is sometimes called the wavebreaker, and contains the machinery for control of the ship. Superimposed on this again is a conning tower with two periscopes.

From this it will be seen that the immersed surface causing frictional resistance is very much greater in the submerged condition than in the floating one. In addition to this, the wave-making resistance of the wave breaker, conning tower and periscopes is a very considerable amount, and these causes taken together have the effect in nearly all submarines of making the submerged resistance greater by a good deal than that of the ship on the surface, for all practical speeds, although at very high relative speeds the resistance in the floating condition rises so rapidly that it becomes much greater than the submerged resistance. We may recall a most interesting paper on this subject which was delivered to the Institution of Naval Architects in 1911 by Mr. Mason S. Chace, in which the author gave an account of experimental model basin tests carried out for him in the United States Experimental Basin at Washington. Four models were tested, three of them being 12 feet long and 12 inches broad, the draughts were 11¼ inches and the total depth from keel to top of superstructure 15 inches; the floating displacement was 380 pounds, which corresponds to about 300 tons for a vessel whose dimensions were represented by the model on a scale of 1 inch to 1 foot. For the vessels corresponding to these models the reserve of buoyancy was about 22 per cent. of the total submerged displacement, and the wetted surface when totally submerged was 54 per cent. greater than the wetted surface when floating. In one of the models where the form of superstructure was varied, the increase was 73 per cent. Without going into the details of variation made, it will be seen at once by how great a proportion the surface is increased, and consequently the frictional resistance, which is a large part of the total resistance at low speeds. It was also clearly demonstrated that within the limits of the experiments the submerged resistance decreased with an increase of the depth to which the model was submerged, but that even when the model was run at a depth which submerged the deck about 28 inches, the effects of surface disturbance or wave making were not eliminated. The amount of increased wetted surface when submerged, as compared with that when floating is greater than would at first sight appear to be warranted by the difference in the respective displacements, but it must be remembered that the displacement available when submerged is confined to the pressure hull, whereas the wetted surface is that of the non-watertight superstructure. In some vessels part of this non-watertight portion is available for surplus buoyancy when in the floating condition, being supplied with valves and scuppers for that purpose, but this buoyancy is not available when the vessel is submerged, for the structure is too light to bear the pressures encountered when diving. It is, however, of great service in giving to the submarine good seagoing qualities on the surface. The curves of resistance as given by Mr. Chace for the floating condition and for the awash condition are similar in form, that of the awash condition being higher than that of the floating condition, because the vessel then has both greater surface friction and greater wave making than when in the floating condition. Up to a speed length ratio of $\frac{V}{\sqrt{L}} = .8$ the curves are approximately fair, but at a speed of $\frac{V}{\sqrt{L}} = 1$ there is a marked hump, followed by a hollow, while at $\frac{V}{\sqrt{L}} = 1.25$ the resistance runs up so rapidly as to put such speeds out of court for practical purposes. The curve of resistance for the submerged condition is free of humps, and is higher than that of the floating condition all the way until a speed $\frac{V}{\sqrt{L}} = 1.3$ is reached, so that it may be said that for all the running conditions the power required when

submerged is greater than that when on the surface for the same speed. The ratio existing between these powers, depends to a very large extent upon the type of vessel, shape and size of superstructure and erections, whether bow rudders are fitted, what kind of keels and after rudders the vessel has, and to what depth she is submerged, so that no exact figure of comparison can be stated. It is obvious that although these vessels are capable of sustaining the pressure at depths of about 100 feet, their general submerged depth for surface will be much nearer the surface, in order to make use of the periscopes for navigation purposes, so that little practical use can be made of the reduction of resistance which appertains to the greater depth.

From a strategical point of view it is not probable that much advantage can be gained from the attainment of the highest submerged speeds that are practically possible, for it has to be remembered that there is in submarines a double system of propelling machinery, usually Diesel engines for surface, and motor drive by means of electric storage batteries for submerged conditions. It is therefore impossible to augment one system without doing so at the expense of the other, and it seems the best distribution of the weight available for propelling machinery to give the vessel the maximum of surface speed, in order to enable her to "get there," and to provide a quite moderate submerged speed, sufficient to enable her to deliver a torpedo while undetected by the enemy. Probably for these vessels a maximum submerged speed of 10 or 11 knots is ample, the remaining available weight being used in providing a surface speed of 16, 18 or even 20 knots, if the dimensions of such vessels are increased, as it is likely they will be in the near future. The limit to useful speed, when submerged, will possibly soon be reached, but the surface speed may be augmented with great benefit to the attacking value of the vessel. Mr. Chace gives the following example of dimensions and powers for such a vessel from the curves of resistance, assuming 50 per cent. efficiency:—Length on load line, 232 feet; surface displacement, 1,200 tons; submerged displacement, 1,536 tons; approximate power to attain 10 knots submerged, 660 indicated horsepower, and for 20 knots on the surface, 6,280 indicated horsepower; but he considers that these figures could be improved upon with the same displacement. It is clear that the development of the submarine depends to some extent upon the development of the internal-combustion oil engine, for the Diesel type of engine has proved itself so suitable that it has no serious rival. Long non-stop runs are continually being made, and now that the reliability of the larger sizes of these engines is assured, there seems little to prevent a great expansion in submarine design.—"The Engineer."

THE OIL CONSUMPTION AND MEAN EFFECTIVE PRESSURE OF DIESEL ENGINES.

By R. Royds, M. Sc., and J. W. Campbell, M. Sc.

It is commonly taken for granted that the total oil consumption of a Diesel engine is a straight-line function of the power developed. No doubt in some cases, where the number of tests at different loads have been few in number, it appears to be so, but an analysis of a large number of tests made on different engines shows that the oil consumption, plotted on a horsepower base, gives in general a distinct curve for each engine. This is clearly indicated by the results of tests on nine different engines, shown in Figs. 1 to 9; particulars of the engines are given in Table I.

TABLE I.—DIESEL ENGINES.

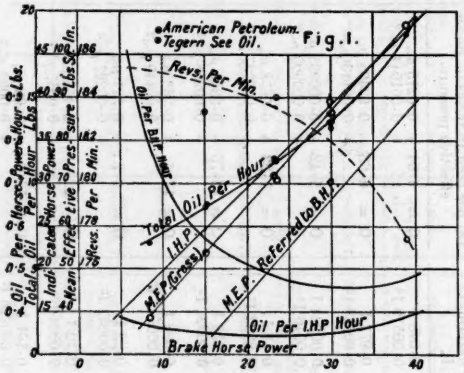
Reference figure.	Maker.	Cylinder dimensions. Dia., in.	Cylinder dimensions. Stroke, in.	Piston speed at full load, ft. per min.	Compressor, particulars and dimensions.	Name of oil used.	Calorific value B.Th.Us. per lb.	Names of experimenters.	Reference.
Fig. 1	M.A.N.	11·8	18·1	550	Single cylinder, 1·9in. dia. and 3·1in. stroke	American petroleum and raw Tegern See oil	About 18,540 (net)	Prof. Eugene Meyer	See article by Bryan Donkin, THE ENGINEER, April 26th, 1901
Fig. 2	—	11·8	18·1	540	Takes air from motor cylinder at 150 lb. per sq. in., further compression in single cylinder compressor, 1·57in. dia. and 3·54in. stroke	Texas crude oil	19,080 (gross) 17,900 (net)	E. G. Hiller	See "Practical Engineer," September 25th, 1903
Fig. 3	—	35 brake horse power, dimensions not given			—	Crude Texas oil	19,150 (gross)	Ade Clark	See "Proceedings," Institution of Mechanical Engineers, 1903, and THE ENGINEER, August 22nd, 1902
Fig. 4	M.A.N.	11·8	18·1	575	Single cylinder, 2·05in. dia., 3·62in. stroke	Petroleum	17,700	Chr. Eberle	"Zeitschrift des Bayerischen Revisions-Vereins," 1906
Fig. 5	Mirrlees	12	18·25	610	L.P. cylinder, annular space between, 5¼in. and 12in. dia., H.P. 1¾in. dia., stroke 8·5in.	Pumpherston gas oil	18,000 (net)	Royds and Campbell	—
Fig. 6	Sulzer	3 cylinders 15	22	570	Air compressor for each cylinder, 1·18in. dia. and 1·97in stroke	Galician crude petroleum residue	18,100	Prof. Gustav Weber	"Schweizerischen Bauzeitung," vol. 50, No. 26
Fig. 7	M.A.N.	2 cylinders 17·72	26·8	715	L.P. cylinder 5¼in., H.P. cylinder 2¾in., stroke 8in.	Gas oil	17,700	Chr. Eberle	"Zeitschrift des Bayerischen Revisions-Vereins," 1906
Fig. 8	Sulzer	3 cylinders 17·8	26	695	—	Galician crude oil	18,100 (gross)	J. Cochand and M. Hottinger	"Zeitschrift des Vereins Deutscher Ingenieure," 1912, page 458
Fig. 9	Carels (?)	3 cylinders 22·05	29·53	740	Air compressor driven by an electric motor, cylinders 8·79 sq. in., 32·3 sq. in., and 102·4 sq. in., stroke 7·087in.	Galician crude oil	Four values given from 18,144 to 20,049 (net)	M. Longridge	"Chief Engineer's Report for 1904, British Engine, Boiler, and Electrical Insurance Company, Limited"
Fig. 12	Nobel	4 cylinders 13·0	14·9	870	Two-stage air compressor, L.P. 9·05in. dia., H.P. 3in. dia., stroke 7·1in.	Crude naphtha	19,600	Selliger	"Zeitschrift des Vereins Deutscher Ingenieure," 1911, vol. 55, April 15th
Fig. 13	M.A.N.	4 cylinders. full load at 400		Normal B.H.P. 300 revs. per min.	—	Galician gas oil	18,120	Chr. Eberle	"Zeitschrift des Bayerischen Revisions-Vereins," 1908, No. 1

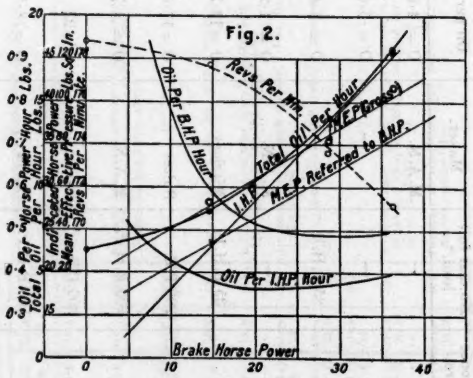
TABLE II.

Reference figure.	Normal full load.*	Maker.	Oil per hour in terms of I.H.P.	Oil per hour in terms of mean effective pressure.	Relation of I.H.P. to B.H.P.
Fig. 1	—	M.A.N.	$O = 4{\cdot}04 + 0{\cdot}048\ I + 0{\cdot}00519\ I^2$	$O = 4{\cdot}04 + 0{\cdot}0215\ p_m + 0{\cdot}001044\ p_m^2$	$I = 0{\cdot}97\ (B + 10{\cdot}4)$
Fig. 2	—	—	$O = 6{\cdot}63 - 0{\cdot}077\ I + 0{\cdot}00728\ I^2$	$O = 6{\cdot}63 - 0{\cdot}03405\ p_m + 0{\cdot}00143\ p_m^2$	$I = 1{\cdot}04\ (B + 7{\cdot}4)$
Fig. 3	35 B.H.P.	—	$O = 5{\cdot}42 + 0{\cdot}00224\ I + 0{\cdot}0046\ I^2$	$O = 5{\cdot}42 + 0{\cdot}000995\ p_m + 0{\cdot}000905\ p_m^2$	$I = 0{\cdot}92\ (B + 17{\cdot}7)$
Fig. 4	—	M.A.N.	$O = 1{\cdot}23 + 0{\cdot}18\ I + 0{\cdot}0032\ I^2$	$O = 1{\cdot}23 + 0{\cdot}0858\ p_m + 0{\cdot}000727\ p_m^2$	$I = 0{\cdot}925\ (B + 13{\cdot}5)$
Fig. 5	35 B.H.P.	Mirrlees	$O = 13{\cdot}75 - 0{\cdot}453\ I + 0{\cdot}00994\ I^2$	$O = 13{\cdot}75 - 0{\cdot}2355\ p_m + 0{\cdot}00268\ p_m^2$	$I = 0{\cdot}993\ (B + 17{\cdot}45)$
Fig. 6	200 P.S.	Sulzer	$O = 17{\cdot}6 + 0{\cdot}096\ I + 0{\cdot}000585\ I^2$	$O = 17{\cdot}6 + 0{\cdot}263\ p_m + 0{\cdot}00449\ p_m^2$	$I = 1{\cdot}02\ (B + 61{\cdot}7)$
Fig. 7	200 B.H.P.	M.A.N.	$O = 4{\cdot}7 + 0{\cdot}22\ I + 0{\cdot}00037\ I^2$	$O = 4{\cdot}7 + 0{\cdot}521\ p_m + 0{\cdot}00207\ p_m^2$	$I = 1{\cdot}05\ (B + 49{\cdot}6)$
Fig. 8	300 P.S.	Sulzer	$O = 22 + 0{\cdot}12\ I + 0{\cdot}0004\ I^2$	$O = 22 + 0{\cdot}453\ p_m + 0{\cdot}0057\ p_m^2$	$I = (B + 82)$
Fig. 9	500 B.H.P.	Carels (?) ...	$O = 13{\cdot}8 + 0{\cdot}161\ I + 0{\cdot}000226\ I^2$	$O = 13{\cdot}8 + 1{\cdot}03\ p_m + 0{\cdot}0093\ p_m^2$	$I = 1{\cdot}105\ (B + 116)$ †

* See Table I. for cylinder dimensions. † Air blast compressor was independently driven, and in this case B = engine brake horse power – power to drive compressors.

Upon consideration, it would appear to be a reasonable relation, for it is well known that there is much difficulty in getting complete combustion of the oil fuel at light loads, when there is a large excess of air for the combustion of the oil, and also at heavy loads, when the excess of air is comparatively small.* From this curve relation it follows naturally that there must be some most economical load† for each engine. In each figure from 1 to 9 is shown the oil consumption per indicated horsepower hour and per brake horsepower hour, all plotted on a brake horsepower base, and it is seen that in general these curves show minimum values at some point below maximum load. These curves have been derived in each case from the mean line drawn to represent the total oil consumption.





* The curve relation between oil consumption and power is also indicated by the theoretical oil consumption of the Diesel cycle with incomplete expansion, when plotted on a theoretical power base, but this cannot account for the actual relation at light loads or at heavy loads.

† In this article the term "most economical" is intended to refer solely to the point of lowest fuel consumption per horsepower hour.

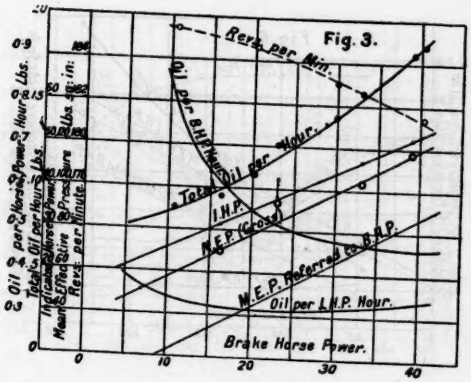


Fig. 3.
Revs. per Min.
Oil per B.H.P. Hour.
Total Oil per Hour.
I.H.P.
M.E.P. (Gross)
M.E.P. Referred to B.H.P.
Oil per I.H.P. Hour.
Brake Horse Power.

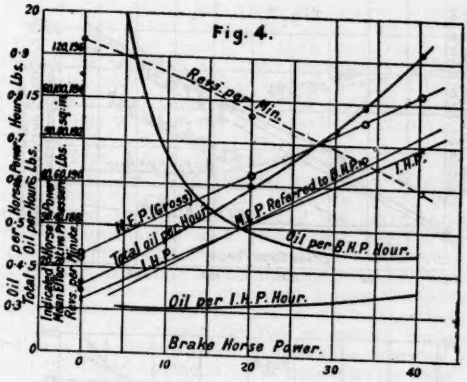


Fig. 4.
Revs. per Min.
M.E.P. (Gross)
Total oil per Hour
I.H.P.
M.E.P. Referred to B.H.P.
I.H.P.
Oil per B.H.P. Hour.
Oil per I.H.P. Hour.
Brake Horse Power.

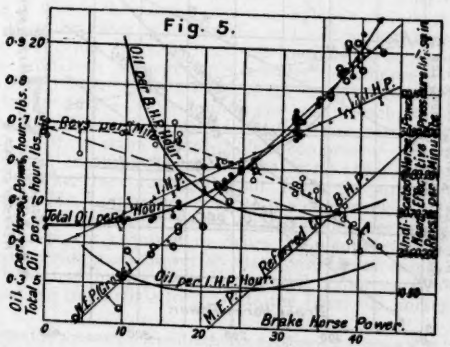


Fig. 5.
Oil per B.H.P. Hour.
Revs. per Min.
I.H.P.
Total Oil per Hour
B.H.P.
Oil per I.H.P. Hour.
M.E.P. Referred to B.H.P.
Brake Horse Power.

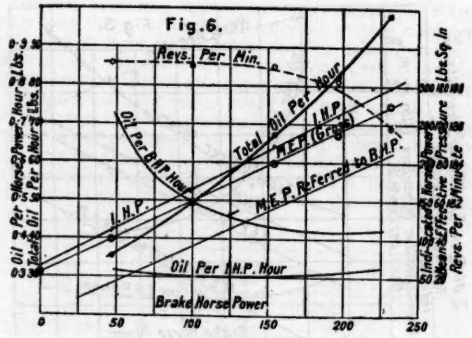


Fig.6.
Revs. Per Min.
Total Oil Per Hour
I.H.P.
M.E.P. (Gross)
Oil Per B.H.P. Hour
M.E.P. Referred to B.H.P.
I.H.P.
Oil Per I.H.P. Hour
Brake Horse Power
0
50
100
150
200
250

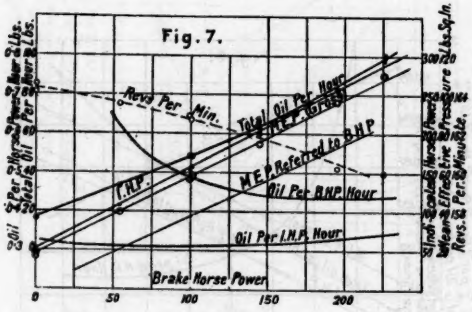


Fig.7.
Revs Per Min.
Total Oil Per Hour
M.E.P. (Gross)
M.E.P. Referred to B.H.P.
I.H.P.
Oil Per B.H.P. Hour
Oil Per I.H.P. Hour
Brake Horse Power
0
50
100
150
200

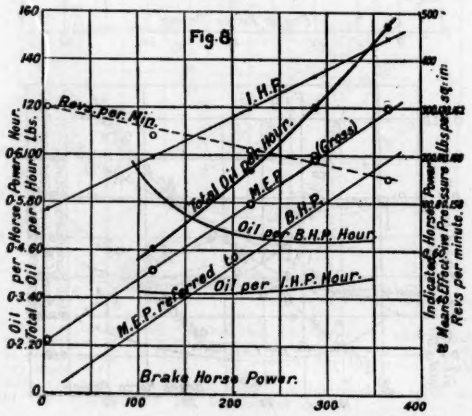


Fig.8.
I.H.P.
Revs. Per Min.
Total Oil per Hour.
M.E.P. (Gross)
B.H.P.
Oil per B.H.P. Hour.
M.E.P. referred to
Oil per I.H.P. Hour.
Brake Horse Power.
0
100
200
300
400

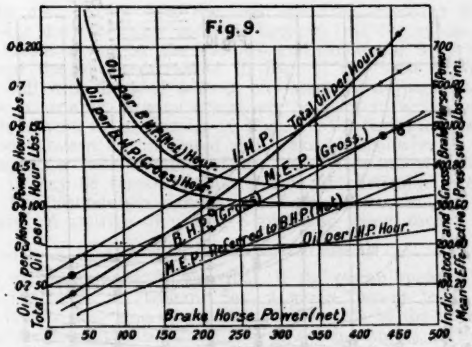
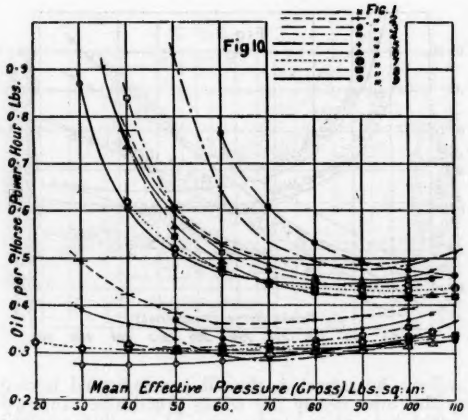


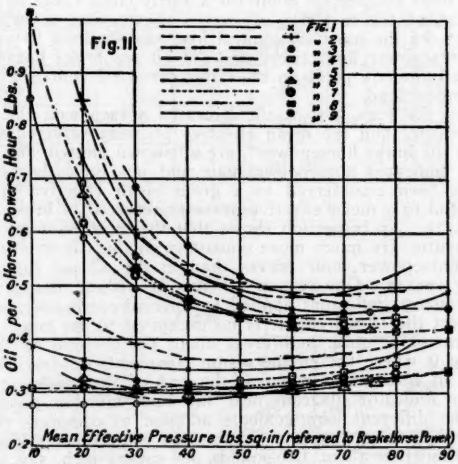
Fig. 9.
Oil per B.H.P. (Net) Hour
Oil per B.H.P. (Gross) Hour
I.H.P.
Total Oil per Hour
M.E.P. (Gross)
B.H.P. (Gross)
M.E.P. Referred to B.H.P. (Net)
Oil per I.H.P. Hour
Brake Horse Power (net)
Total Oil per Hour Lbs.
Oil per Brake Horse Power Hour Lbs.
Indicated and Gross Brake Horse Power
Mean Effective Pressure Lbs. per Sq. In.
0 50 100 150 200 250 300 350 400 450 500

As would perhaps be expected, the oil per indicated horsepower hour curve is very flat over nearly the whole practicable range of load, and the oil per brake horsepower hour curve shows little variation on each side of the most economical point for a fairly large range of load, whilst a comparison of the two sets of curves shows what a large difference there is between the most economical loads as obtained from the oil per indicated horsepower hour curve and the oil per brake horsepower hour curve. The minimum point on the latter curve, of course, gives the true most economical load.

In each of the Figs. 1 to 9, the indicated horsepower, the gross mean effective pressure and the mean effective pressure referred to or corresponding to the brake horsepower*, are all shown plotted, and by their aid the oil per indicated horsepower hour and per brake horsepower hour curves have been transferred to a gross mean effective pressure base in Fig. 10, and to a mean effective pressure referred to brake horsepower base in Fig. 11. An inspection shows that the most economical mean effective pressures are much more consistent among themselves in the oil per brake horsepower hour curves than in the oil per indicated horsepower hour curves. One reason for this is without doubt the unreliability of ordinary indicators applied to internal-combustion engines, and in this respect the Diesel engine is no exception to the rule. Also, some builders seem to be quite indifferent about the position of the indicator hole, placing it where the engine piston completely covers it up, or even overruns it in some cases when at the end of the stroke, consequently affecting the indicator diagram near the fuel injection point. Another reason is the different arrangements adopted to compress the blast or fuel-injection air, some using a single-stage compressor taking air from the main cylinder at about 150 pounds per square inch, say, and driving the compressor by links and levers from the main piston. Others use a two-stage or a three-stage water-cooled compressor driven from the engine either by a crank on the end of the crank shaft or by levers from the piston, in either case taking the air from the atmosphere. In the engine tested by Mr. Longridge—Fig. 9—the compressors were independently driven by electric motors, and in estimating the net or resultant brake horsepower, the power required to drive the compressors was deducted from the engine brake horsepower so that approximate comparisons might be made with engines having directly-operated compressors. Little or no attempt is made in this article to separate the frictional horsepower of the

* *i. e.*, Mean effective pressure referred to brake horsepower = gross M.E.P. $\times \frac{\text{B.H.P.}}{\text{I.H.P.}}$.





engine from that required to operate the compressors. Attempts in this direction have been made by indicating the air compressors, but no reliance can be placed on the indicator diagrams taken from small high-pressure compressors running at high speeds. When the blast compressor is independently driven, and the power required to drive it is measured, this is likely to exceed the actual power required when driven direct by the engine. Besides these considerations, unless the throttle valve at the air inlet to the compressor is altered with the load on the engine, the blast pressure falls as the load decreases, because the fuel valve is usually arranged to open for a definite period of the cycle, which is not varied with the load, and the less the quantity of oil to be injected into the

cylinder the more easily does the blast air find access to the cylinder and the lower does the blast pressure become. As far as can be judged from the values given by the various experimenters, in practically all cases, except the one represented in Fig. 5, the blast pressures were allowed to fall with the load without any apparent adjustment or regulation. This, of course, is what commonly happens in actual practice. In any case, it is seen by inspection of Figs. 1 to 9 that there is a straight line relation between the indicated and brake horsepowers.

Returning to a consideration of the most economical mean effective pressures, it may be remarked that this is of particular interest to the designer. To illustrate the significance of this factor, let p_m mean effective pressure. A = area of piston, S = piston speed, then

$$\text{Indicated horsepower} = kp_mAS \quad . \quad . \quad . \quad . \quad (1)$$

when k is a constant. It follows that, if the piston speed S is selected, the product p_mA is a constant for a given power to be developed. Thus, the higher the value of p_m the smaller will be the necessary value of A, and therefore the smaller the cost of the engine. Therefore, the designer would desire to adopt the highest possible mean effective pressure at the normal load of the engine, consistent with reliability, and with the fulfilling of any guarantee as to capacity for overload without an excessive sacrifice of economy. An inspection of the curves shows that the economy varies but little between 80 pounds and 110 pounds per square inch gross mean effective pressures. It is not desirable to attempt to obtain a gross mean effective pressure much beyond 110 pounds per square inch, as the excess of air for combustion in the cylinder is then comparatively small, and the cooling of large cylinders and valves sufficiently becomes a difficult problem. The following approximate calculations illustrate the magnitude of the excess air for combustion:—Cylinder diameter, 12 inches; stroke, 18.25 inches; gross mean effective pressure, 110 pounds per square inch; revolutions per minute, 200; indicated horsepower = 57.3; brake horsepower = 40, say; oil used per cycle, 0.00314 pound; air temperature at beginning of compression, say, 120 degrees F.; air theoretically necessary per pound oil, say, 14.6 pounds; volume of air per pound at 120 degrees F. and 14.7 pounds per square inch, 14.6 cubic feet; volume swept by piston per stroke, 1.2 cubic feet; weight of air taken in per cycle $= \frac{1.2}{14.6} = 0.082$ pounds. Then,

$$\text{Ratio} \frac{\text{Air supplied}}{\text{Air theoretically necessary}} = \frac{0.082}{0.00314 \times 14.6} = 1.79 \quad . \quad . \quad (2)$$

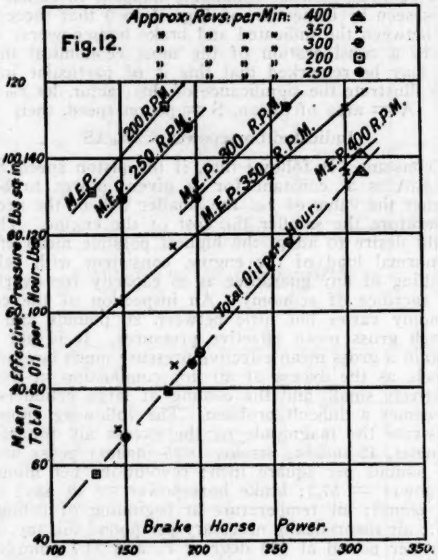
In the appendix it is shown that, to a first approximation, oil per hour =

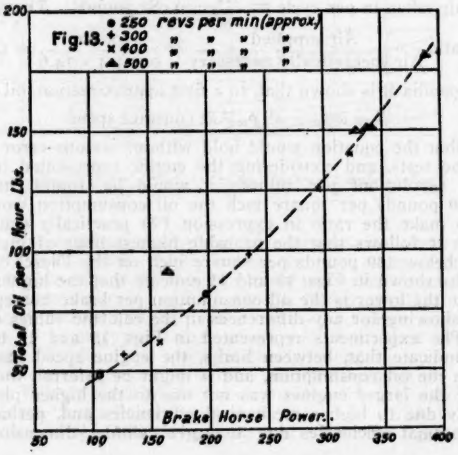
$$b + a^1 p_m + c^1 (p_m)^2 \text{ at constant speed} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Assuming that the equation would hold without serious error beyond the range of the tests, and considering the engine represented in Fig. 5 to run at 200 revolutions per minute, it would be found that when p_m is about 140 pounds per square inch the oil consumption would then be sufficient to make the ratio in expression (2) practically equal to unity, from which it follows that the probable highest limit of mean effective pressure is below 140 pounds per square inch on the Diesel cycle.

The results shown in Figs. 10 and 11 indicate that the higher the power of an engine the lower is the oil consumption per brake horsepower hour, even after allowing for any differences in the calorific values of the various oils. The experiments represented in Figs. 12 and 13 by the total oil curves indicate that, between limits, the engine speed has not much influence on the oil consumption, and it might be inferred that the great economy of the larger engines was not due to the higher piston speeds, but probably due to higher mechanical efficiencies and, perhaps, slightly increased thermal efficiencies due to larger cylinder dimensions.

Referring now more particularly to the author's experiments represented in Fig. 5, it will be seen that numerous tests were made, and these extended for a period of over two years. Special short tests were made in addition to determine the influence of the blast pressure on the oil consumption, the procedure being to count the revolutions required to





consume a definite quantity of oil, say, 500 cubic centimeters, with the brake load constant and the blast pressure varied. It was found that the blast pressure did not influence the oil consumption appreciably except when too low or too high to give the normal type of diagram, and that a lower blast pressure could be used more economically at light loads than at heavy loads.* Normally the blast pressure was arranged at 650 pounds to 700 pounds per square inch at full load for the gas oil used, down to about 600 pounds per square inch at light loads, the compression pressure being about 500 pounds per square inch. It is seen from Fig. 5 that the total oil consumption comes out practically constant at all loads below about 10 brake horsepower. Mr. Ade Clark referred‡ to the same tendency in the engine he tested, though this is not shown in Fig. 3, as he did not obtain any values himself below the lowest load shown. It will also be noticed that the average speed curve in Fig. 5 is practically constant over a large range of load, and the authors are of opinion that the governor is much too sensitive at light loads. This over-sensitiveness causes a decided, though stable, hunting of the engine and governor at loads below about 11 brake horsepower, and not only was this hunting evident from the running of the engine, but also from the erratic indicator diagrams below this load. At no load a series of diagrams could be obtained varying from zero up to nearly full-load mean effective pressure. Various blast pressures were tried frequently, and also various jacket temperatures, but the hunting seemed to be independent of any such circumstances.* The only other reference to hunting at light loads which the authors have come across was in the experiments represented in Fig. 2, and it was borne out by the variable indicator diagrams at no load.

To ascertain more definitely whether the blast pressure had any appreciable influence on this hunting effect, the authors obtained a series of speed diagrams on a Horne tachograph, and a close inspection indicated that the hunting effect dies out between 9.3 and 12.5 brake horsepower, and is hardly affected by the blast pressure.* The hunting is quite stable, and would hardly cause much trouble in ordinary installations having mechanical driving, but it might have a serious influence if the engines were driving alternators in parallel. Owing to the engine being required for other experiments, the authors could not investigate this hunting effect further, but they would suggest that it is more common in such engines than is generally admitted. From Figs. 1 to 9 the average speeds are plotted for each set of tests, and though the small number of points available make the curves doubtful in several cases, they would indicate that only in one or two instances is over-sensitiveness at light loads apparently avoided. It may be said as a general statement that the speed power relation ought to be more nearly a straight line like A B in Fig. 5. Modifications of the governor or the links connecting the governor sleeve to the fuel-controlling valve could be made so as to reduce the sensitiveness at light loads without reducing the overall sensitiveness from full load to no load. Such a possible modification is suggested in Fig. 14, where the full lines represent the common arrangement for controlling the fuel supply from the governor, and the heavy dotted lines represent a possible modification, reducing the sensitiveness at light loads and increasing it slightly at heavy loads, but retaining the same average sensitiveness from no load to full load. It may be noted, however, that in this modification the frictional resistances at the various pins would cause a greater reactive force on the governor sleeve than in the usual

* More recent experiments made on this engine show that there may be conditions when the blast pressure appeciably affects both the hunting and oil consumption at light loads, but the conditions producing this are so obscure that no satisfactory explanation has so far been obtained. In any case, it behooves buyers of Diesel engines to insist on oil consumption, and governing guarantees at or near no loads as well as for higher loads.

‡ "The Engineer," 22d August, 1902.

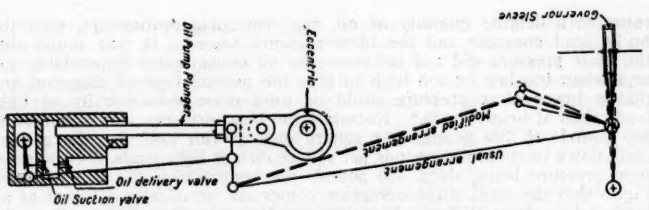


Fig. 14.

arrangement. In any case, where the governor is too sensitive at light loads, there is every indication that this is likely to cause hunting at these loads with an augmentation of the necessary oil consumption, though it may not have any appreciable influence on the oil consumption at or near full load.

Appendix.

The relations between the total oil consumption and the indicated and brake horsepowers can be expressed by equations of the following forms:—

If W = total oil per hour, pounds,
I = indicated horsepower of engine,
B = brake horsepower,
p_m = gross mean effective pressure, pounds per square inch, the equation for the straight-line relation between I and B can be expressed as

$$I = k\ (B + m), \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where k and m are constants. It will be seen from Table II that k is not far from unity. Again, the general relation between W and I is given by

$$W = b + a\,I + c\,I^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where a, b and c are constants for each engine and the particular oil used.

Assuming a standard constant speed for each engine it is evident that equation (2) could be modified to

$$W = b + a'\,p_m + c'\,p_m^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The values of the constants in equations (1), (2), and (3) are quoted in Table II. Further, from (1) and (2),

$$W = b + a\,k\,(B+m) + c\,k^2\,(B+m)^2 \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Again, since

$$\frac{W}{I} = \frac{b}{I} + a + c\,I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Then differentiating,

$$\frac{d\left(\frac{W}{I}\right)}{dI} = -\frac{b}{I^2} + c.$$

And this is zero when

$$I^2 = \frac{b}{c} \text{ or } I = \sqrt{\frac{b}{c}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Signifying that the oil consumption per indicated horsepower hour is a minimun when $I=\sqrt{\frac{b}{c}}$.

Also

$$\frac{W}{B}=\frac{b}{B}+\frac{a\,k(B+m)}{B}+\frac{c\,k^2(B+m)^2}{B}$$

$$=\frac{b+a\,k\,m+c\,k^2m^2}{B}+a\,k+2\,c\,k^2m+c\,k^2B$$

differentiating,

$$\frac{d\left(\frac{W}{B}\right)}{dB}=-\frac{(b+a\,k\,m+c\,k^2m^2)}{B^2}+c\,k^2$$

and $\frac{W}{B}$ will be a minimun when $d\frac{\left(\frac{W}{B}\right)}{Bd}=O$,

i. e., when

$$B^2=\frac{b+a\,k\,m+c\,k^2m^2}{c\,k^2}$$

or when

$$B=\sqrt{\frac{b+a\,k\,m+c\,k^2m^2}{c\,k^2}} \quad . \quad . \quad . \quad . \quad . \quad (7)$$

or taking $k=I$ when

$$B=\sqrt{\frac{b}{c}+\frac{a\,m}{c}+m^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

which thus represents the brake horsepower at the point of minimum oil consumption per brake horsepower hour.

A comparison of the various values of a, b and c for the different engines shows that considerable variations occur, but it is significant that in the two cases to which Figs. 2 and 5 refer the value of a comes out negative, and in both these cases the engine hunted at light loads. It is interesting to note, also, that when a is negative, and supposing the law $W=b+a\,I+c\,I^2$ could hold down to zero indicated horsepower, the total oil consumption would come to a minimum value before attaining zero indicated horsepower. In fact, in the particular case represented by Fig. 5, there is indication of a minimum value at some power above zero brake horsepower.—"The Engineer."

MARINE MOTOR NOTES.

The battle of the two-cycled and the four-cycled oil engine is becoming as exciting as the battle of the boilers was a few years ago, when both from an engineering and a commercial point of view advocates of the water-tube boiler met the defenders of the Scotch boiler in argument, armed with data of a more or less convincing quality. The "Paragon" type of engine, recently brought out, promises to make itself a character in keeping with its name, according to the description of its cycle of operations and the method pursued in its design and setting, with a view to obtain as full a value as possible out of the fuel. The designer and joint patentee is an enthusiast and is ever on the move to effect economies by seeking to obtain the highest calorific value out of the prime mover

and the highest propeller efficiency by the introduction of a reducing agent, electrical and mechanical, but with a decided leaning to the former. The "Paragon" engine is designed for either the two or the four-cycle, aiming in each case at the highest thermal efficiency. Our attention has been called to another design which has been the outcome of much patient labor and may shortly be introduced when it has been further developed. The recent trials of the two-cycle motor vessel *Hagen* for oil carriage, built by Messrs. Krupp for the German-American Petroleum Co., has shown results of a very satisfactory character in the power developed and the speed attained. The vessels now in course of building for different companies, added to those which are under consideration, will make quite a large amount of tonnage. The Atlantic transport vessels will be a most interesting trio, oil fuel for producing steam in boilers; oil fuel for internal-combustion engines, and coal for producing steam. The engines in each case of the three vessels will be the most efficient present-day types. The *Juno, Selandia* and *Christian X* and other vessels fitted with oil engines appear to be giving good accounts of themselves, and we note that in the course of the adjourned discussion on Mr. Zulver's paper, published last month, it was stated that the *Juno* had been running experimentally and successfully on taracan oil, a sample of which was shown to the meeting of the Institute of Marine Engineers and was described as an unrefined or crude oil. Formerly solar oil was in use in order to give the engines every chance of showing what they could do under the most favorable conditions before attempting economies in quality of fuel supplies. The keen controversies which have been evoked as to the available quantities of coal and oil in the world to meet the growing requirements, with the probabilities as to prices and relative values, have also had the effect of rekindling the possibilities with regard to the distillation of a suitable spirit from vegetable tissue, cultivated to meet the demands of the market, thus relieving to some extent the increasing prices. The question of the duty upon such a distillation is one which has many side issues, ranging from the pleasant paths dedicated to the luxuries of the comparative few to the wants or even necessities of the many, and this latter issue comes into consideration when cost of transit or locomotion affecting the carriage of goods and produce has to be carefully gone into.—"The Marine Engineer and Naval Architect."

FOREIGN DREADNOUGHT DEVELOPMENTS.

With the exception of the United States, France is the only naval power of any standing which announces the characteristics of her new warships in advance of their completion. In the majority of cases, however, the details leak out by some means or another, and in only a very few instances are the main features of large ships now under construction unknown.

The most striking point in recent dreadnought development is the increasing size and power of the gun. The British navy was the first to abandon the 12-inch gun as the main armament of dreadnoughts and four battleships are now in commission—*Orion, Thunderer, Conqueror* and *Monarch*—which have ten of these weapons, firing 1,250-pound projectiles in five center-line turrets. The same caliber of gun is mounted in the four ships of the *King George V* class, and the four of the *Iron Duke* class, now completing; but the gun is of a much more powerful type. The length of the shell has been increased and 200 pounds added to its weight, much of the addition being accounted for by an increased weight of bursting charge. Another notable development in British dreadnought designs is the restitution of the 6-inch gun.

In view of the interest taken in America in the subject of the battle cruiser, it is important to note that this type is now being "abandoned" in England. That, at any rate, is the way in which the latest development of the capital ship is generally described; but it would be equally true to say that the battleship is being abandoned and that in the future only battle cruisers will be built. The essential difference between the two types is that the battleship has thicker armor, more guns and less speed than the battle cruiser. Thus, to take two contemporary ships, the battleship *Iron Duke* has a 12-inch belt, carries ten 13.5-inch guns, and is designed for 22.5 knots, while the *Tiger* battle cruiser has a 9¾-inch belt, carries eight 13.5-inch guns, and is designed for 28 knots. The first four battle cruisers built for the British navy were designed for 25 knots, and the *Iron Duke* class are the first battleships to be designed for more than 21.

Last year four large armored ships were included in the British program —*Queen Elizabeth, Barham, Valiant* and *Warspite*—and these will represent the amalgamation of the two types. Their designed speed is 25 knots, which, if past experience goes for anything, will mean a trial speed of 27 knots. They are to carry only eight big guns, but these will be of 15-inch caliber, firing 1,950-pound projectiles; the total broadside will be 15,600 pounds, and their main belts will, it is believed, be 15 inches thick. It will be seen, therefore, that they will have the characteristic battle cruiser armament of eight big guns, as well as a very high speed, while the only battleship feature remaining to them will be the thick armor belt. It is noteworthy that in these ships the disposition of the *Michigan's* guns will be reproduced.

The German authorities have not been very happy with their dreadnought designs. The first eight battleships built have twelve big guns each (11-inch in the first and 12-inch in the second four), but only eight in either case are available for broadside fire. In the *Kaiser* and *Friedrick der Grosse,* recently completed, the number of 12-inch is reduced to ten, but there is a considerable gain in efficiency, inasmuch as all can fire on either beam, though only over a small arc, as two of the turrets are placed *en echelon.* The next group, those of the *Kaiserin* class, have twelve 12-inch, and although it is understood there is a full broadside, the arrangement of the turrets is not known. It was not until the 1st of March last that the first German ship carrying a heavier gun than the 12-inch was launched. This was the *König,* a 27,000-ton ship, which will have ten Krupp 14-inch guns, firing 1,367-pound shells in five center-line turrets, the total broadside being 13,670 pounds.

From the point of view of design, the most interesting foreign dreadnoughts now projected are undoubtedly the French battleships *Normandie, Gascogne, Flandre* and *Languedoc,* which are to be laid down this year. These ships will be 574 feet long and 88.5 feet in beam, with a displacement of 25,200 tons, a designed speed of 21 knots, and a maximum thickness of 12.6 inches of side armor. Their main armament will comprise twelve 13.4-inch guns, representing a total broadside of 15,984 pounds, and these will be mounted in three quadruple turrets on the center-line, there being four guns abreast in each turret. There will also be twenty-four 5.5-inch guns and six submerged torpedo tubes. The three-gun turret is now a well-established feature in many navies—it has been adopted by the United States, Italy, Austria and Russia; but the interesting experiment about to be made by the French authorities will be watched with interest.

The Brazilian battleship *Rio de Janeiro,* now completing at Armstrong's works at Newcastle, makes a striking contrast to these French vessels. Originally intended to carry twelve 14-inch guns on a tonnage of 32,000, the *Rio de Janeiro* was redesigned at the desire of the Brazilian authorities after she had been some months in hand, and is now to carry fourteen 12-inch on a displacement of 27,500 tons, the total broadside

being 14,900 pounds. The guns will be distributed over seven turrets, all mounted on the middle line—two forward (one superposed), two amidships between the funnels, and three aft (the center one superposed). The length of the ship is 670 feet over all, and the designed speed 22 knots, but the main armor belt is only 9 inches thick. The anti-torpedo battery consists of twenty 6-inch guns.

Italy has always been noted for originality in warship design. The *Dante Alighieri,* carrying twelve 12-inch on 19,400 tons, was the first triple-turreted warship to be completed, and that nation has now in various stages of construction five ships which are to carry thirteen 12-inch. Ten of these are arranged in the same manner as the 14-inch of the *Nevada* and *Oklahoma,* and there is an additional three-gun turret amidships. The American arrangement referred to is reproduced exactly in the *Morosini* and *Dandolo,* laid down in December, but the Italian vessels will differ in having less armor, a designed speed of 25 knots, and a displacement of 28,000 tons. A striking advance is to be made in the case of two ships to be laid down this year. For these vessels the Admiralty had under consideration two designs, one showing a ship of 28,000 tons carrying nine 15-inch in three turrets, and the other a vessel of 35,000 tons carrying twelve guns of this caliber in four turrets, 25 knots being the speed in ease case. The latter design is stated to have been decided upon. Given another 800 tons these ships would be exactly twice as heavy as the *Dreadnought* of 1906. The Italians favor big ships, and their latest, if they be built, will surpass anything building or afloat, their total broadside of 23,400 pounds exceeding even that of our own *Pennsylvania,* which is 16,800 pounds. She can scarcely equal the *Pennsylvania* in protection, since a large part of her big displacement must be allotted to the powerful motive power necessary to drive her at 25 knots. —" Scientific American."

FLUXES FOR USE IN MELTING SOFT METALS.

When the so-called soft metals, such as tin, lead, zinc and their various alloys, are melted, the surface of the molten metal oxidizes and produces what is known as " dross." Dross is the metallic oxide mixed with more or less metal. It is not all oxide, as many persons believe, but the oxide becomes entangled in the metal itself, producing a mass which becomes thick and does not flow readily.

It is the function of fluxes when soft metals are melted to act upon this dross (or in reality, the oxide in it) and clean and clear the surface of the metal.

Those who are not familiar with the action of fluxes should try an experiment and be convinced of their value. Melt some lead and it will be noticed that the surface is covered with a film. If the lead is scrap material, then a greater quantity of dross will form, for the reason that the scrap lead is already oxidized to some extent and this oxide floats on the surface of the metal. Now throw on a small piece of rosin. The rosin will immediately melt and clear the surface so that it will have a bright appearance and as clean as the surface of pure water. The rosin dissolves the oxide, forming a slag which floats to the side of the ladle in which the lead has been melted, and may readily be skimmed off.

The use of fluxes in melting soft metals is quite important, particularly when scrap metals are used, as they serve to cleanse the molten metal from oxide and produce clean, clear material that will readily run into molds. Scrap metals, as a usual rule, melt with the formation of considerable dross, and if no flux is used on them the bars of metal cast will not be clean. By the use of fluxes, however, it is possible to obtain nearly as clean metal as though new materials had been employed.

The question as to what kind of flux to use in melting soft metals will depend upon the metal itself. The following directions may be safely followed:

Tin and Its Alloys.—This list comprises the solders, britannia metal, block tin, babbitt metals with a tin base and any other alloys of which tin is the principal constituent.

For tin and its alloys *rosin* is the best flux, as it requires but a small quantity on the surface after melting to clean it. *Sal-ammoniac*, too, is a good flux for tin and seems to work better if the metal is used quite hot, and at a temperature at which the rosin burns off. The rosin, however, is to be preferred, as it is free from the fumes of sal-ammoniac.

Lead and Its Alloys.—This class comprises some of the solders, which contain lead in excess, babbitt metals with a lead base, pewters, and the various antimonial-lead alloys, together with pure lead itself, and other various alloys containing lead as the principal ingredient. For lead and its alloys *rosin* is the best flux. It acts in a manner similar to that of tin and at once clears the surface. Sal-ammoniac does not work well on lead or its alloys unless quite hot, and then it seems to have a good effect. This action, however, is not satisfactory when the lead is at the heat ordinarily used for pouring metals, and the temperature at which it does act is so high that it is rarely employed in practice. The rosin, therefore, is the best flux for use in melting lead and its alloys.

Zinc and its Alloys.—There are but few zinc alloys used, of which zinc is the principal constituent, and one of the chief components of this class is the well-known die castings. The white brasses (some of the white brasses contain tin as the principal ingredient) are also made with zinc as the chief constituent. Zinc itself, however, is extensively used. The best and in fact the only flux that can be used on zinc and its alloys is *sal-ammoniac*. This has an excellent cleansing action and produces clear, clean metal from that which is quite dirty. Rosin is not suitable for use on zinc and its alloys, and although it has some cleansing action, it is not satisfactory. Sal-ammoniac, however, seems to give all that is desired in the way of a flux for zinc and its alloys.

The manner of using the fluxes is simple. After the metal is melted, whatever it may be, tin, lead or zinc, or their alloys, the flux is thrown on the surface and stirred in. No particular amount of flux can be given for use, as it will depend upon the amount of dross to be cleaned from the surface. The way to do is to throw on a small quantity of the flux, stir and skim off the dross as clean as possible. Then add more, if required, until the surface is clean. By adding the flux little by little, instead of a large quantity at one time, the progress can be watched, and it is not wasted, for the reason that only just enough is used to clean the metal.

The fluxes are always used on the surface after the metal is melted, and art *not* put in while the melting is taking place.—"The Brass World."

NEW PUBLICATIONS OF THE BUREAU OF MINES OF THE DEPARTMENT OF THE INTERIOR.

BULLETINS.

Bulletin 48. Selection of explosives used in engineering and mining operations, by Clarence Hall and S. P. Howell. 1913. 50 pp. 3 pls., 7 figs.

Bulletin 55. The commercial trend of the gas producer in the United States, by R. H. Fernald. 1913. 92 pp. 1 pl. 4 figs.

Bulletin 62. National Mine-Rescue and First-Aid Conference, Pittsburgh, Pa., September 23–26, 1912, by H. M. Wilson. 1913. 74 pp.

TECHNICAL PAPERS.

Technical paper 38. Wastes in the production and utilization of natural gas, and means for their prevention, by Ralph Arnold and F. G. Clapp. 1913. 29 pp.

Technical paper 48. Coal-mine accidents in the United States, 1896–1912, with monthly statistics for 1912, compiled by F. W. Horton. 1913. 72 pp., 10 figs.

The Bureau of Mines has copies of these publications for free distribution, but cannot give more than one copy of the same bulletin to one person. Requests for all papers cannot be granted without satisfactory reason. In asking for publications please order them by number and title. Application should be addressed to the Director of the Bureau of Mines, Washington, D. C.

WAR SHIP TONNAGE OF THE PRIN

NUMBER AND DISPLACEMENT OF WAR SHIPS, BUILT AND BUILDING, OF 1,500 OR MORE T

TYPE OF VESSEL.	GREAT BRITAIN. §				GERMANY.				UNITED STATES.				FRANCE.			
	BUILT.		BUILDING.		BUILT.		BUILDING.		BUILT.		BUILDING.		BUILT.		BUILDING.	
	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.
Battleships* (Dreadnought type)	18	324,250	11	201,100	11	226,480	6	136,220	8	167,680	5	140,000			7	161
Battleships** (Pre-dreadnought)	40	888,980			20	343,980			24	225,858			20	296,060		
Coast-defense vessels†					3	12,300			4	12,960			2	15,400		
Battle cruisers††	8	160,800	3	85,000	5	64,264	3	81,000								
Armored cruisers	24	406,000			9	94,345			11	149,200			20	201,720		
Cruisers‡	74	978,595	14	78,425	30	130,900	4	19,062	35	73,700			10	69,925		
Torpedo boat destroyers	144	86,721	40	36,276	116	63,642	12	8,023	42	28,177	14	14,891	78	32,647	8	8
Torpedo boats	69	11,486			9	1,600			19	3,706			157	15,370		
Submarines	70	26,166	16	13,400	26	11,740	6	5,000	23	6,191	23	11,630	75	27,265	20	12
Total tons built and total tons building		2,997,247		476,200		865,984		267,894		768,132		166,219		687,787		173
Total tons built and building		3,473,545				1,133,878				929,351				807,717		

NOTES APPLYING TO THE TABLES OF THE WARSHIP TONNAGE OF THE PRINCIPAL NAVAL POWERS.

* Battleships having a main battery of all big guns. (11 inches or more in caliber.)

** Battleships of (about) 10,000 or more tons displacement, whose main batteries are of more than one caliber.

† Includes smaller battleships and monitors.

†† Armored cruisers having guns of largest caliber in main battery and capable of taking their place in line of battle with the battleships. They have an increase of speed at the expense of carrying fewer guns in main battery and a decrease in armor protection.

‡ All unarmored warships of more than 1,500 tons are classed as cruisers. Scouts are considered as cruisers in which battery and protection have been sacrificed to secure extreme speed. The word "protected" has been omitted because all cruisers except the smallest and oldest now have protected decks.

§ Colonial vessels included.

The following vessels are not included in the tables: Ships over twenty years old from date of launch, unless they have been reconstructed and re-armed within five years; torpedo craft over fifteen years old; those not actually begun or ordered, although authorized; transports, colliers, repair ships, torpedo depot ships, or other auxiliaries; vessels of less than 1,500 tons, except torpedo craft; torpedo craft of less than 50 tons. Vessels undergoing trials are considered as completed. The tonnage for ships building has been estimated.

RELATIVE ORDER OF WARSHIP TONNAGE.

Present order (tonnage completed).

Nation.	Tonnage.
Great Britain	2,007,247
Germany	865,984
United States	763,132
France	627,787
Japan	471,962
Russia	286,930
Italy	224,837
Austria	198,159

As would be the case if vessels now building were completed.

Nation.	Tonnage.
Great Britain	2,483,545
Germany	1,133,878
United States	929,351
France	807,717
Japan	616,528
Russia	595,807
Italy	413,882
Austria	260,751

E PRINCIPAL NAVAL POWERS.

OFFICE OF NAVAL INTELLIGENCE,
NAVY DEPARTMENT.

F 1,500 OR MORE TONS, AND OF TORPEDO CRAFT OF MORE THAN 50 TONS.

FRANCE.			JAPAN.				RUSSIA.				ITALY.				AUSTRIA.			
LT.		BUILDING.	BUILT.		BUILDING.		BUILT.		BUILDING.		BUILT.		BUILDING.		BUILT.		BUILDING.	
Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.
.....	7	161,884	3	61,000	1	20,000	.....	.....	7	188,600	1	19,000	7	187,300	2	46,000	2	40,000
206,005	.....	.....	15	191,300	..	.....	6	112,000	.....	.....	5	56,100	.....	.....	6	74,613	.....	.....
13,400	.....	.....	2	8,986	.....	.....	2	10,200	.....	.....	.....	.....	.....	.....	6	41,700	.....	.....
.....	.....	.....	.....	.....	6	110,000	.....	.....	4	120,000	.....	.....	.....	.....	.....	.....	.....	.....
200,736	.....	.....	15	238,488	..	.....	6	62,500	.....	.....	9	74,000	.....	.....	3	23,000	.....	.....
49,025	.....	.....	14	60,200	.....	.....	7	52,845	2	5,000	5	13,000	3	10,720	4	19,202	3	19,440
32,647	6	5,150	20	122,983	2	1,200	26	20,575	9	9,450	24	8,667	11	7,910	12	4,720	6	4,720
18,570	.....	.....	34	5,450	.....	.....	14	2,120	.....	.....	45	5,305	21	2,415	40	6,500	15	2,000
27,362	20	12,000	13	2,672	2	1,200	31	6,645	8	4,277	16	5,055	2	840	6	1,656	7	4,472
627,787	...	179,000	...	671,963	.....	144,600	.....	266,900	.....	306,827	.....	284,027	.....	208,040	.....	199,190	.....	69,632
806,787			696,638				266,927				613,063				268,781			

SEA STRENGTH OF THE POWERS.

The Office of Naval Intelligence, Navy Department, under date of April 1, 1913, publishes the interesting data here given as to the sea strength of the chief navies of the world. This information has been compiled, it is stated, in order to answer the frequent inquiries of societies and persons throughout the country interested in the maintenance of the Navy and its relative strength and importance in comparison with foreign navies.

PERSONNEL.

Rank.	England.	France.	Germany.	Japan.	United States.
Admirals of the Fleet	3	.....	2	1	1(*a*)
Admirals	12	.....	5	7	.....
Vice Admirals	22	15	12	17	.....
Rear Admirals	55	30	21	45	24(*g*)
Captains and Commanders	644	360	351	292	212
Other line officers	2,473	1,467	1,811	1,818	1,471
Midshipmen at sea	558	60	398	154	0
Engineer officers	872	505	529	683	.....
Medical officers	593	390(*c*)	322	368(*f*)	317
Pay officers	685	211	259	341	221
Chaplains	139	.....	28	.....	23
Warrant officers	2,675	2,445(*d*)	2,615	1,520	697
Enlisted men	115,079	55,760	60,920	42,043	47,469
Marine officers	457	.....	170	.....	316
Enlisted men (Marines)	20,943(*b*)	.....	5,826(*e*)	.....	9,866
Total	145,210	61,243	73,269	47,289	60,617

(*a*) The Admiral of the Navy.
(*b*) Includes 3,130 men of the Coast Guard.
(*c*) Includes pharmacists and apothecaries.
(*d*) Includes adjutants, premier maitres and maitres of all branches.
(*e*) Marine infantry and seaman artillery.
(*f*) Includes pharmaceutical officers.
(*g*) The United States now has, temporarily, as extra numbers, due to promotion for war service, and to officers restricted by law to engineering duty only on shore only, 8 flag officers, 23 captains, 6 commanders, 10 lieutenant commanders and 1 lieutenant.

—"Army and Navy Journal."

SHIPS.

PROGRESS OF NAVAL VESSEL CONSTRUCTION.

The Bureau of Construction and Repair, Navy Department, reports under date of May 10, 1913, the following degree of completion of vessels building for the United States Navy:

No.	BATTLESHIPS.	Knots.		Degree of completion. Per cent. Apr. 1,'13.	May 1, '13.
34	*New York*	20½	Navy Yard, New York	75.20	77.20
35	*Texas*	21	Newport News S. B. Co.	86.90	88.40
36	*Nevada*	21	Fore River S. B. Co.	25.30	29.30
37	*Oklahoma*	20½	New York S. B. Co.	20.50	26.00
38	*Pennsylvania*	...	Newport News S. B. Co.	...	...
	TORPEDO-BOAT DESTROYERS.				
43	*Cassin*		Bath Iron Works	85.00	87.20
44	*Cummings*		Bath Iron Works	77.90	81.20
45	*Downes*		New York S. B. Co.	41.80	47.50
46	*Duncan*		Fore River S. B. Co.	71.20	79.00
47	*Aylwin*		Wm. Cramp & Sons	89.80	92.10
48	*Parker*		Wm. Cramp & Sons	86.90	89.20
49	*Benham*		Wm. Cramp & Sons	83.00	87.90
50	*Balch*		Wm. Cramp & Sons	85.40	88.50
51	*O'Brien*		Wm. Cramp & Sons	0.70	1.30
52	*Nicholson*		Wm. Cramp & Sons	0.70	1.20
53	*Winslow*		Wm. Cramp & Sons	0.70	1.20
54	*McDougal*		Bath Iron Works	3.30	4.40
55	*Cushing*		Fore River S. B. Co.	5.90	8.90
56	*Ericsson*		New York S. B. Co	3.00	5.00
	SUBMARINE TORPEDO BOATS.				
23	F-4		Electric Boat Co., Seattle	96.00	100.00
26	G-4		American Laurenti Co., Philadelphia	88.30	88.40
27	G-2		Lake T. B. Co., Bridgeport	86.60	87.20
28	H-1		Electric Boat Co., San Francisco	88.90	90.40
29	H-2		Electric Boat Co., San Francisco	88.90	90.00
30	H-3		Electric Boat Co., Seattle	87.00	88.00
31	G-3		Lake T. B. Co., Bridgeport	65.10	67.40
32	K-1		Electric Boat Co., Quincy	70.90	76.60
33	K-2		Electric Boat Co., Quincy	69.20	76.80
34	K-3		Electric Boat Co., San Francisco	73.60	75.60
35	K-4		Electric Boat Co., Seattle	72.20	75.00
36	K-5		Electric Boat Co., Quincy	55.20	58.70
37	K-6		Electric Boat Co., Quincy	54.40	57.10
38	K-7		Electric Boat Co., San Francisco	62.30	65.30
39	K-8		Electric Boat Co., San Francisco	60.40	63.40
40	L-1		Electric Boat Co., Quincy	...	...
41	L-2		Electric Boat Co., Quincy	...	...
42	L-3		Electric Boat Co., Quincy	...	...
43	L-4		Electric Boat Co., Quincy	...	...
44	L-5		Lake T. B. Co., Bridgeport	...	...
45	L-6		Lake T. B. Co., Long Beach, Cal.	...	...
46	L-7		Lake T. B. Co., Long Beach, Cal.	...	...
47	M-1		Electric Boat Co., Quincy	...	...

No.			Degree of completion. Per cent. Apr. 1.	May 1,
		SUBMARINE TENDER.		
1	*Fulton*	New London S. & E. B. Co., Quincy	10.60	11.60
		FUEL SHIPS.		
9	*Proteus*	Newport News S. B. Co	87.30	90.70
10	*Nereus*	Newport News S. B. Co	80.60	87.10
12	*Jason*	Maryland Steel Co	98.80	98.80
	Jupiter	Mare Island Navy Yard	96.80	99.10
13	*Kanawha*	Mare Island Navy Yard	...	...
14	*Maumee*	Mare Island Navy Yard	...	...
		GUN BOATS.		
19	*Sacramento*	Wm. Cramp & Sons	9.00	11.10
20	*Monocacy*	Mare Island Navy Yard	25.70	32.70
16	*Palos*	Mare Island Navy Yard	25.70	32.70

BRAZIL.

NEW BRAZILIAN BATTLESHIP.

The main particulars of the Brazilian battleship *Rio de Janeiro* are as follows: Length between perpendiculars, 632 feet; length over all, 668 feet; breadth, 89 feet; draught, 27 feet; displacement (about), 27,500 tons; speed, 22 knots; normal coal supply, 1,500 tons; full coal supply, 3,000 tons.

The *Rio* will have the huge complement of about 1,100 officers and men.

The main armament will consist of fourteen 12-inch guns, twin mounted in armored turrets. The seven turrets are all on the center line of the ship, and consequently all the 12-inch guns can be trained on one broadside simultaneously. On no other warship up to the present can so many big guns be brought to bear on one broadside, and it is probable that this arrangement represents the last word in 12-inch gunned ships, as it does not seem likely that any increase beyond fourteen will be contemplated. The broadside fire of the 12-inch guns of the *Rio* will be 11,900 pounds, while the ahead and astern fire will be 3,400 pounds.

The two forward turrets are on the forecastle deck, the second being superposed. The two turrets amidships are also on the forecastle deck, placed back to back, and separated only by the boat derrick. The remaining three turrets are placed on the upper deck, aft, the middle one being superposed.

The secondary armament will consist of twenty 6-inch guns, ten 3-inch guns, and three 21-inch submerged torpedo tubes. Fourteen of the 6-inch guns will be disposed in single positions, along the upper-deck battery behind 6-inch armor, and the remaining six 6-inch guns will have shields. The main armor is to be 9 inches thick in the center, tapering down to 4 inches at the ends, and 9 inches is also the thickness of the 12-inch gun barbette armor. There is a conning tower forward, with armor 12 inches thick.

The machinery will consist of Parsons steam turbines, working four screws, with Babcock & Wilcox boilers, the furnaces of which are fitted to burn oil fuel as well as coal.—"A. and N. Register."

BRITISH EMPIRE.

THE NAVY ESTIMATES, 1913–14.

The First Lord's statement, introducing the Estimates, was delivered on March 26th. A special feature was the proposal for a "naval holiday"—a halt for one year in building and bettering. "If," said the First Lord, "for the space of a year, for 12 calendar months, no new ships were built by any nation, in what conceivable manner would the interests of any nation be affected or prejudiced? You have good ships today. They are the best in the world—till better ones are built. Can they not have at least one year's reign before they are dethroned? Why should we not take a naval holiday, for one year, so far, at any rate, as new construction of capital ships is concerned?"

Interesting features in the statement were the proposal by the Admiralty to form an Imperial Squadron, composed entirely of ships presented by the Colonies, to be based on Gibraltar within easy reach of all parts of the Empire; the decision to retain coal for the present as the main motive power of the British line of battle; the policy of the Admiralty towards aviation; the effect of the withdrawal of the Marconi company from the wireless contract; and the steps now being taken to arm our merchantmen, a necessity forced on us by the action in that direction of certain of the great Powers.

The shipbuilding policy pursued is in accordance with that outlined in the House in July last, the 60 per cent. margin being maintained and, in addition, two capital ships being laid down to every additional one of Germany. "Additional to this total will be any ships which we may have to build in consequence of new naval developments in the Mediterranean." The ships of the Colonies are to be over and above this reckoning. Amongst other details outlined were the organization and composition of our battle fleet, a comparison being effected with that of Germany.

The Navy Estimates for 1913–14 were presented to Parliament on March 13th and issued as a Parliamentary Paper (1). In his explanatory statement, accompanying the Estimates and also presented to Parliament [Cd. 6668], the First Lord of the Admiralty stated:

"The Estimates for 1913–14 amounted to £46,309,300, as compared with £45,075,400 for the current year (including the Supplementary Estimate). The principal increases occur under the heads of Pay and Personnel (Vote I), Victualling and Clothing (Vote II), and Naval Armaments (Vote IX). The increase in the vote for personnel is due mainly to the requirements of new ships now being placed in commission and under construction. It is proposed to reach a total of 146,000 officers and men by March 31st, 1914. This requires an addition to Vote A of 8,500 and an increase in average bearing throughout the year of 5,000. The maximum figure is given for Vote A instead of, as heretofore, the average figure, which was somewhat misleading. The increase of £712,200 in Vote I is due chiefly to the pay required for the additional personnel, and to meet the charge for increases of pay to officers and men recently approved. The increase under Vote II is to provide for victuals and clothing for additional numbers of the fleet and for the maintenance of reserves and stores. The increase in the armaments vote is mainly due to the requirements of new construction. The increases under Vote XII and the non-effective votes are mainly automatic, the addition to the non-effective charge being £91,900. The extraordinary pressure of work in the shipyards and the scarcity of labor are leading to short earnings by contractors on the continuation programs; and I do not estimate that more than £11,224,000 will be spent on this branch of new construction (Vote VIII) within the year as compared with £12,067,727 estimated for 1912–13. Every effort will be made to secure punctual deliveries, and should conditions change and progress improve, a further estimate will

be presented later in the year. A sum of £2,052,400 is required for beginning work on ships of the new program, which is composed as follows:

"Five battleships, eight light cruisers, 16 destroyers, together with a number of submarines and subsidiary craft. The total cost of the new program is £15,958,525, as compared with £13,014,000 in 1912–13."

Appended to the explanatory statement is a statement of work done during the past year, of which the following are the main features:

SHIPBUILDING.

Between April 1st, 1912, and March 31st, 1913, the following ships will have been completed and become available for service:

Battleships—*King George V; Centurion; Thunderer; Conqueror.*

Battle cruisers—*Lion; Princess Royal; New Zealand.*

Light cruisers—*Chatham; Dublin; Southampton; Amphion; Melbourne* (for the Commonwealth of Australia).

Destroyers—*Attack; Badger; Lizard; Hydra; Goshawk; Phœnix; Firedrake; Lurcher; Oak; Beaver; Acasta; Christopher; Shark; Achates; Cockatrice.*

Submarines—*D.6; E.1; E.4.*

Miscellaneous—*Maidstone; Adamant; Alecto.*

On April 1st, 1913, there will be under construction: Eleven battleships (including *Malaya*); three battle cruisers (including one for the Commonwealth of Australia); 13 light cruisers (including one for the Commonwealth of Australia); 35 torpedo-boat destroyers; 21 submarines (including two for the Commonwealth of Australia); a number of vessels for carrying oil fuel, and for various fleet services.

NEW CONSTRUCTION.

The *Conqueror* and *Thunderer* have been completed and commissioned. The *King George V* has been completed and commissioned, and the *Centurion*, which has been delayed owing to a collision during the early stages of her steam trials, will be completed in April. The *Ajax* and *Audacious* have both been launched. The *Iron Duke* and *Marlborough* of the 1911–12 program have been launched, and progress made with the *Delhi* and *Benbow*, the two other battleships of this program. The four battleships of the 1912–13 program, *Queen Elizabeth, Warspite, Barham* and *Valiant*, have been laid down, the two former at Portsmouth and Devonport, and the two latter at the yards of Messrs. John Brown & Company and the Fairfield Company at Glasgow respectively.

Of the battle cruisers, the *Princess Royal* and the *New Zealand* (built for the New Zealand Government) have been completed and commissioned, and the *Queen Mary* will shortly proceed on trial, and prepare for completion and commission. Progress has been made with the battle cruiser *Tiger* of the 1911-12 program at the works of the Messrs. John Brown & Company.

Of the five vessels of the *Chatham* class under construction, the *Chatham, Dublin* and *Southampton* have been completed and commissioned; H.M.A.S. *Melbourne* has been completed and commissioned for the service of the Australian Commonwealth. The remaining vessel, H.M.A.S. *Sydney*, is expected to have completed her steam trials and be commissioned early in the new financial year. A cruiser of this type, H.M.A.S. *Brisbane* is being built in Australia by the Commonwealth Government. Detailed drawings have been supplied by the Admiralty for the use of the Commonwealth authorities, and tenders for machinery are being invited in this country. The three light cruisers, 1911–12, are under construction, the *Birmingham* at the works of Messrs. Armstrong, Whitworth & Company, Ltd.; the *Lowestoft* at H. M. Dockyard, Chatham; the *Nottingham* at H.M. Dockyard, Pembroke. Substantial progress has been made on all these ships; the *Lowestoft* will be launched on April

23d next. The orders for the eight light cruisers of the 1912–13 program have been placed as follows: One at H.M. Dockyard, Chatham; one at H.M. Dockyard, Devonport; three with Messrs. Beardmore & Company; two with Vickers, Ltd.; one with the Fairfield Shipbuilding Company. All these ships are to be completed in the summer of 1914. The light cruiser *Amphion* is being completed at H.M. Dockyard, Pembroke, and will probably be commissioned by the end of the present financial year. The light cruiser *Fearless*, also under construction at Pembroke, was launched in June last, and should be completed early in the new financial year.

All the destroyers of the 1910–11 program have been delivered and are in commission. Two of the destroyers of the 1911–12 program have been delivered and three more are expected to be delivered before the close of the present financial year. The twenty vessels of the program for 1912–13 have been ordered, and good progress has been made in their construction.

Good progress has been made with the construction of submarines. The depôt ship for submarines, the *Maidstone* and her tenders, the *Adamant* and *Alecto*, have been completed.

The surveying ship *Endeavour* has been completed and delivered. It has been decided not to proceed with the two shallow-draught steamers, *Kingfisher* and *Rail*, for service in China, until the new financial year. The *Woolwich*, depôt ship for torpedo-boat destroyers, is nearing completion. The two floating docks for Portsmouth and the Medway have been completed and delivered; a small floating dock for destroyers has been completed and placed at Harwich; and one for submarines, to be stationed ultimately at Dover, has been completed and is at present at Sheerness. A small floating dock for destroyers has been ordered for Portland for completion next year. The S.S. *Knight Companion*, *Tabaristan*, and *Heliopolis* (to be renamed *Mediator*) have been provisionally purchased, subject to satisfactory trials and survey, and will be converted into a fleet repair ship, a depôt ship for torpedo-boat destroyers, and a hospital ship respectively.

ORGANIZATION OF THE FLEET.

The Home, Atlantic and Mediterranean Fleets have been reorganized. The battleships in full commission (at present 29 in number, to be raised later to 33) have been organized in four battle squadrons of the 1st Fleet, with a fleet flagship. Corresponding to these are the 1st Battle-Cruiser Squadron and the 2d, 3d and 4th (armored) Cruiser Squadrons; the last, having only recently been constituted out of the Training Squadron, is for the time being manned on a 2d Fleet basis. In addition to the 1st Fleet the Home Fleets comprise a 2d Fleet, with a 5th and 6th Squadron (the latter still in process of formation), and a 3d Fleet, with a 7th and 8th Battle squadron and six more cruiser squadrons. The 2d Fleet is manned with active (full nucleus) crews and the 3d Fleet with reserve (reduced nucleus) crews. For the present, however, the 6th Cruiser Squadron will be manned on a 3d Fleet basis.

The Mediterranean Fleet will in future comprise a battle cruiser squadron (styled the 2d) of four ships of the *Indomitable* class and an armored-cruiser squadron (the 1st) also composed of four powerful ships.

To the 1st Fleet are attached four fully-manned destroyer flotillas, and a fifth will be formed as new destroyers are delivered. Four other destroyer flotillas manned with active crews have been organized as a separate command under an "Admiral of Patrols," and are known as "Patrol Flotillas." In the same organization are comprised the bulk of submarine flotillas. The growing importance of this arm has been recognized by the advancement of the officer in charge of the submarine service to commodore.

The preliminary sea training of boys for the fleet will in future be carried out in cruisers of the *Edgar* class based on Queenstown, in charge of a captain. These vessels have special complements in peace, but will belong to cruiser squadrons of the 3d Fleet on mobilization.

The organization of the remaining squadrons and detached ships has not been varied in any important respect.

WORK OF THE FLEET.—A general survey of work of the Navy follows, details being given of the maneuvers, general service abroad of the fleet, and the ceremonies and visits of both fleets and individual ships during the year.

NAVAL BASES AND WORKS.—Progress on the important naval works under construction at Rosyth has been interfered with by strikes and labor difficulties, but every effort is being made by the contractors to recover the lost ground. The construction of a third dock has been ordered. Permanent moorings have been laid for destroyers using the anchorage to the west of the Forth Bridge. In the Humber an oil-fuel depôt is under construction for the Admiralty, with a view to the river being used as a base for torpedo craft. Contracts have also been let for the erection of oil-fuel storage in the Medway, at Invergordon and at Portsmouth. The inner harbor and depôt for destroyers and submarines at Dover is making good progress. The main Admiralty Harbor has been principally used by torpedo craft during the past year.

It is intended to station four battleships and three cruisers of the 3d Fleet at Pembroke. These vessels will be maintained there on the ordinary 3d Fleet basis. As already stated, Queenstown will in future be used as a base for boys' training cruisers, eight vessels being detailed for this service. By the stationing of these fifteen vessels at Pembroke and Queenstown it is anticipated that the difficulty which has been felt for some time past in finding berthing room at the three principal naval ports will be reduced.

PERSONNEL.

The committee appointed to inquire into the education and training of cadets and midshipmen, presided over by Admiral Sir Reginald Custance, has presented its report, which is now under the consideration of the Board. It has been found possible, however, to make certain changes recommended without waiting for a decision on the more important questions involved. The examinations in seamanship and in navigation and pilotage are now passed at sea after two years' and four months' service as midshipmen, instead of on shore after three years as hitherto, and, on passing, midshipmen are rated as acting sub-lieutenants. The remaining examinations, in gunnery, torpedo and engineering, are held eight months afterwards, but the examination in voluntary subjects has been abolished. The annual examination of midshipmen afloat has also been abolished.

As regards the War Staff, development has proceeded on the lines laid down in the memorandum published last year; thirty-nine naval and seven marine officers have been appointed to form the nucleus of the War Staff. The first War Staff course, to which twelve naval and three marine officers were appointed, commenced in April last and has just terminated. A second class, consisting of twelve naval and three marine officers, began the course at the end of last month. A series of lectures on international law and prize manual, merchant shipping and court-martial procedure, supplemented by instruction in the principles of strategy and tactics, has been started at the Branch War College at Devonport, and the reports on the first two courses are satisfactory. Owing to the expansion of the work at the Branch War College at Chatham, a captain has been appointed to take charge of it.

A new scheme was introduced last autumn to enable warrant officers, petty officers and seamen to reach the rank of commissioned officer at

an early age. The candidates selected undergo courses of instruction at Portsmouth, and on passing are given the rank of acting mate. They then proceed to the Royal Naval College at Greenwich for four months' instruction in navigation, followed by two months' instruction in pilotage at the Navigation School at Portsmouth. On passing the examination at the termination of this course, they are confirmed as mates and are embarked in sea-going ships for two years, at the end of which time they are eligible for promotion to the rank of lieutenant. Their duties as lieutenants will be the same as those of other lieutenants, and they will be considered for promotion to commander with other lieutenants on their merits.

Other points touched on were the relative rank and age of entry of officers in the Royal Marines; the pay of executive officers; the loan of officers to the Australian Government; the Naval Medical School at the Royal Naval College, Greenwich. As regards the Training Service, additional temporary accommodation has been provided for the harbor training of boys by the transfer of youths, formerly trained in H.M.S. *Ganges II* at Harwich, to Devonport Barracks, and by the addition of H.M.S. *Powerful* to the *Impregnable* establishment at Devonport. Under these arrangements about 1,050 additional boys can be accommodated for harbor training. The inspecting captain of boys' training ships has now been relieved of the command of H.M.S. *Impregnable* and has been appointed for the duty of inspecting and supervising generally the training of boys.

The numbers of recruits raised for 1911–12 were 11,576 naval ratings and 1,556 marines, an increase of 270 over 1910–11; the estimated cost of the increase in pay of officers and men of the royal navy and royal marines works out at £386,473 per annum. Reference is made to the steps taken to combat tuberculosis in the fleet, and to the appointment of a committee to consider the ventilating of warships. The question of the night accommodation of men on week-end leave and the revision of summary punishments receive attention.

Royal Marines.—The numbers borne on March 31st, 1913, were estimated at about 16,300. Changes in the entry of commissioned officers and in promotion from warrant rank are, amongst other points, noticed.

Coast Guard.—The authorized establishment of coast-guard officers and men is 3,130. The numbers borne on January 1st, 1913, were: District captains, district paymasters and staff, 35; divisional officers, 76; chief officers and men, 2,962; total, 3,073.

Royal Fleet Reserve.—The new class of the Royal Fleet Reserve, called the "Immediate" class, has been established, composed of seamen, stokers and marines under 32 years of age who have left the service before completing time for naval pension. These men are enrolled for a period of five years, and are required to perform 28 days' training on board H.M. ships annually, and to undertake to come into actual service if summoned by the Admiralty on a national emergency. They draw a retainer of 1s. a day so long as they fulfil the condition of service in the "Immediate" class, and on completing their service in that class are allowed to re-enrol in Class B. Since the institution of the class last May, 2,180 men were enrolled up to December 31st last, the majority of these being transfers from Class B.

The total numbers of the Royal Fleet Reserve have increased from 24,082 to 25,788, the distribution of these numbers on December 31st, 1912, being as follows:

	Class A.	Class B.	Immediate Class.	Total.
Seamen and Naval police.....	3,714	8,091	902	12,707
Stokers	2,097	5,055	947	8,099
Marines	1,730	2,921	331	4,982
Total	7,541	16,067	2,180	25,788

ROYAL NAVAL RESERVE.—The strength of the Royal Naval Reserve (Home) on January 1st, 1913, was: Officers of the military branch, 1,219; commissioned engineer officers, 171; assistant paymasters, 99; warrant engineers, 158; engine-room artificers, 571; seaman ratings, 10,572; stoker ratings, 5,401. Particulars giving the numbers trained and details of training are added.

ROYAL NAVAL VOLUNTEER RESERVE.—The strength of the force is now six Divisions, comprising 44 companies, the actual numbers being:

Royal Naval Volunteers.	Establishment.	Strength (Jan. 1st, 1913.)
Officers	194	170
Honorary Officers	...	25
Petty officers and men	4,318	3,944
Permanent staff:		
Officers	7	7
Petty officers and men	78	78

It is now under consideration to form another Division on the Forth, and a beginning will be made with three companies of 100 men each, which will be temporarily attached to the Clyde Division.—"Journal of the Royal United Service Institution."

FRANCE.

THE NAVAL ESTIMATES FOR 1913.

The Naval Estimates for 1913 amount to 511,330,537 francs as compared with 457,609,617 francs in 1912 and 415,451,318 francs in 1911. In his introductory remarks, M. Paul Painlevé, the Reporter of the Budget, lays stress on the necessity of providing the dockyards with the proper plant for keeping the fleet in repair, and he draws attention to the works which must be carried out at Toulon and Bizerta if this object is to be attained, with the steps necessary for preserving the preëminence of Brest in view of the concentration of the fleet in the Mediterranean, urgent questions to which attention has been repeatedly drawn during the last two years. He shows that the slowness of promotion in all classes is creating a great deal of discontent, and that certain reforms which have been carried out have been so hastily conceived that carrying them out had only served to discourage the men, especially those who were distinguished more for their general capacity than for their aptitude in passing examinations, and that counsels of prudence were necessary in carrying out certain reforms which had been recommended in the method of recruiting and the "Inscription Maritime." M. Painlevé also states that the problem of powder is still weighing heavily on the Navy and that other questions sink into insignificance alongside it.

He draws attention to M. Delcassé's declaration that no sea has the importance for France that the Mediterranean has, and that although it might be possible to leave the guarding of the Channel to torpedo boats and submarines, this could not be done in the Mediterranean.

"At present the mastery in every sea is only a chimera. Great Britain herself, in spite of her naval power, has been compelled to concentrate almost the whole of her battle units in order to secure supremacy in a limited area, the North Sea. There exists for most nations a maritime zone, the mastery of which is a matter of imperious necessity. For France this zone is the Western Mediterranean. Our country, astride of two seas separated by the immense Iberian peninsula, occupies a particularly disadvantageous strategical position. It is materially impossible for her to reign

at once in the Mediterranean, the Channel or the Atlantic. Nature forbids her an offensive covering, at the same time, both these two regions. If our Navy wishes to take the offensive in one, it must remain on the defensive in the other. We cannot pretend to take the offensive in the North against either of the two great Northern maritime powers. It is, moreover, not necessary that we should seek to be masters in the North Sea; it will be sufficient if we make the navigation in these waters difficult to the enemy and that the Channel and our ocean seaboard should be untenable by their fleets. In the Western Mediterranean, on the other hand, we have the absolute right to maintain our supremacy against all by an energetic offensive, and it is necessary that we should make every sacrifice and use every resource which the exigencies of that supremacy may require.

"It stands thus: the offensive in the Mediterranean, active defensive in the Channel and Atlantic, this is the double objective that the organization of our naval forces should allow us to realize. This is the formula that M. Monis has laid down to the Senate for several years, and is what we have ourselves defended on many occasions before the Chamber. Its acceptance by the Government and the adhesion of Parliament to it compensate us today fully for the attacks which have been made upon us during the two years that we have energetically maintained this conception of our naval policy. They encourage us to persevere on the path we have taken to use every means to obtain the credits agreed to by the country for the development of our battle fleet. We quite understand the perfectly legitimate emotion raised among our brave Western population by this new distribution of our naval forces. We will discuss later the indispensable measures necessary for protecting our great Atlantic war harbor and for even increasing its importance. But the question which dominates all others is to know if the actual general organization of the Navy fully equals at the present time, and if the provisions of the Department for the future are also equal to the realization of the double objective that we have defined."—"Journal of the Royal United Service Institution."

GERMANY.

NEW GERMAN DREADNOUGHT.

On March 1st the new German dreadnought *König* was launched at the Imperial Docks at Wilhelmshaven in the presence of the Emperor. The *König*, which is Germany's fourteenth large battleship, had a weight at launching of 24,000 tons; she surpasses the latest German super-dreadnoughts in size and is consequently the largest battleship hitherto built in a German yard. No details of construction nor dimensions have yet been published, but it is already known that the new battleship will be an improved vessel of the *Kaiser* class. It is also stated that she will be fitted with turbines and will do more than 23.6 knots, which is the speed of the swiftest German dreadnought at present in commission. It may also be taken for granted that a considerable advance will be made in armament and general fighting capacity over the already existing ships. The *König* will be completed by the autumn of next year and will then form part of the High Seas Fleet. Of the twenty battleships forming this fleet, ten are known as "large" battleships. The proportion between these and the battleships of smaller size will be considerably changed before the end of the present year by the completion of the three new dreadnoughts at present nearing completion, viz., the *Kaiserin, Prinzregent Luitpold* and *König Albert*.—"The Engineer."

JAPAN.

JAPANESE WARSHIP ON TRIAL.

The Japanese battle cruiser *Kongo*, the third ship of the dreadnought type to be completed in this country for a foreign Power, has recently completed her steam, gun and torpedo trials. A navigating party of 577 officers and men of the Japanese Navy recently arrived in this country from Japan, under the command of Captain Nikana, I. J. N. They will be on board during the trials, and will take the ship out to Japan on their successful conclusion. The *Kongo* was designed by Messrs. Vickers, and was laid down on January 17th, 1911, and launched on May 18th last year. She is 704 feet long and 92 feet in beam, her displacement at a draught of 27 feet 6 inches being 27,500 tons. Her armament consists of eight 14-inch guns in four center-line turrets, and an anti-torpedo battery of sixteen 6-inch guns. Protection is afforded by a belt of Krupp steel 10 inches thick amidships, tapering to 5 inches forward and 3 inches aft. The main gun barbettes are of 10-inch steel, and there is a protective deck 2½ inches thick. The contract speed is 28 knots with Parsons turbines of 70,000 horsepower, and the maximum fuel capacity is 4,000 tons.—"Page's Weekly."

RUSSIA.

BUDGET.—The efforts being made to regain her naval position are manifest in the large sums set aside for the Navy in the outline of the Russian Budget for 1913. The total of the ordinary and extraordinary expenditure is increased to £340,893,230, of which £24,478,300 goes to the Navy and £57,964,936 to the Army. Last year the navy estimates only came to £17,465,800, and the army estimates to £52,546,186. The navy estimates have thus increased more rapidly than the Army (an increase of about £7,012,500 as compared with about £5,418,750 for the Army).

WIRELESS TELEGRAPHY.—The wireless telegraphy stations at Ochotsk, Najaschan and Amadyr were inaugurated on November 3d, 1912. The wireless telegraphy station at Reval has received new apparatus, which enables it to communicate direct with the Balkan States.

NEW CONSTRUCTIONS: LAUNCHES, ETC.—The two 4,300-ton cruisers (which are not to be confused with the four 6,500-ton cruisers) will be the only exceptions to the principle of building all the vessels of the program in Russia; the reason given is the urgent need of the vessels, which are to serve principally for training the engineer ratings.

The Admiralty would very willingly have bought these vessels ready-built. As no such vessels were for sale, and as the Russian yards asked for too long a period, these two cruisers were ordered from the Schichau yard at Elbing, who have engaged to deliver them by July 15th, 1914. These two vessels will be of the same type as the German *Kolberg*, possess Schichau turbines, develop a speed of 27.5 knots, and cost £371,875 each.

The metropolitan yards have received the following orders: Four armored cruisers (Baltic) are given to the Admiralty yard and the Baltic yard (State Dockyards). The keels were to be laid in 1912, and they are to be completed in four years. Price of the hull, about £2,125,000; price of the armor, turrets, guns, torpedoes and reserve armament, about £2,656,250 each vessel.

Two light cruisers (Baltic) are given to the Putiloff yard. They will have a displacement of 6,500 tons, and cost, without guns, torpedoes and armor, £944,375 each. Two light cruisers (Black Sea) are given to the Nicolaieff Amalgamated yard.

Eight torpedo boats (Baltic) to the Reval Russian Society; six torpedo boats to the St. Petersburg Ironworks; eight torpedo boats to Lange-Becker at Reval; five torpedo boats (Baltic) to the Ziese yard at Reval. The price of each of these torpedo boats, without guns and torpedoes, £212,500, except those constructed in the Ziese yard, which will only cost £207,188.

Six submarinés (Baltic) in the Baltic yards; six submarines by Nobel Lessner at Reval.

PORTS, DOCKYARDS, CANALS.—The surprising thing in the distribution of contracts is that almost all the yards mentioned above do not yet exist. As a matter of fact, an attempt is being made to build up an industry of naval shipbuilding at the same time as a new fleet. The two State shipbuilding yards alone exist, and it will be necessary to complete their enlargement first of all before they can be used in all exigencies. The other yards exist either totally or in part on paper.

The natural result will be that the new constructions will cost very dear, and will be long periods under construction. These drawbacks have been faced and accepted. An attempt has been made to lessen this latter drawback, in the possibility of contracts being badly carried out by entirely new and inexperienced yards, by allowing all the yards to make an appeal for help from abroad as a commencement, and under certain restrictions.—"Journal of the Royal United Service Institution."

ASSOCIATION NOTES.

The following new members and associates have joined the Society since the publication of the last JOURNAL.

MEMBERS.

Hyland, John J., Lieutenant Commander, U. S. N.

Nash, Charles P., Captain of Engineers, U. S. R. C. S.

ASSOCIATES.

Asprea, Pietro, Ing., Cantiere Orlando, Livorno, Italy.

Mower, George A., Managing Director, Sturtevant Engineering Co., Ltd., 147 Queen Victoria Street, London, England.

Soncini, Cesare, Ing., Direttore Cantieri Fluviali & Marittimi, Livorno, Italy.

ERRATA FOR VOL. XXV, NO. 2, MAY, 1913.

Page 183, middle of page:

for $\frac{\text{I.H.P.}_1 \times 132{,}000}{{}_pP_1 \times R_1 \times \pi D_1^2}$, read $\frac{\text{I.H.P.}_1 \times 132{,}000 \times r^2}{{}_pP_1 \times R_1 \times \pi D_1^2}$.

Page 195:

Formula (3), for 101.33, read 101.33 S.

Page 220:

Example for P.A. ÷ D.A., read $\text{P.A.}_{11} \div \text{D.A.}_{11}$

Add P.A. ÷ D.A. . . . . .219, .2525, .2776, .3067.

In formulas for computation at end of article on screw propellers:

Nos. 27 and 29 insert A^4 in the numerator of equation for D; also in Form No. 2 and Form No. 4.

No. 27 should read:

$$\text{D} = \text{Diam.} = \text{(3 blades)} \quad \sqrt{\frac{291.8 \times \text{I.H.P.} \times C^2 \times A^4}{P \times R \times \text{I.T.}_d}}.$$

$$\text{Form No. 2} = \frac{1}{2}\{2.46509 + \log(24) + 2\log(21) + 4\log(3) - \log(17) + \log(8) = \log(27).\}$$

Likewise for No. 29.